

Aalto University
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Measurement of Sound Spectra of Cabinet-built Frequency Converters

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ABSTRACT OF
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<p>This thesis aims to find a method to determine the noise levels for cabinet-built frequency converters that is in accordance with a widely-accepted standard. Two favorable standards are found to determine the sound power level in frequency bands and the total sound power level in engineering grade of accuracy for a frequency converter: ISO 3744 and ISO 9614-2. The former standard relies on the measurement of sound pressure level in discrete positions, and the latter uses sound intensity measurements by scanning the surface of the noise source. The methods in these standards are compared, and it is found that ISO 9614-2 is a more suitable option due to its relaxed requirements on measurement environment.</p> <p>Although frequency converters rarely output noise in harmful levels, a few noise control methods for frequency converters are discussed in theory. Some of these methods may prove valuable, but more extensive study is required before implementing them as a part of a cabinet-built frequency converter. As a result, the weight of the thesis is on the actual measurements to gain reliable noise data that can be further worked on. However, one test case is included where sound absorbing materials are used as a noise control method.</p> <p>Several test cases are defined to produce a set of measurement data on an air-cooled and a liquid-cooled frequency converter. After conducting the measurements and examining the data, it is found that the method defined in ISO 9614-2 is easily repeatable, and that the method also generates reliable data even if it is conducted by a relatively low-experienced person. Finally, further steps are suggested to improve the measurement process.</p>			
Keywords:	frequency converter, drive, noise, noise measurement, noise control, sound intensity, sound intensity measurement, sound power, sound power level, sound pressure, A-weighting		
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<p>Tämä diplomityö käsittelee kaapitettujen taajuusmuuttajien äänimittauksia, ottaen samalla huomioon teollisuudenalan hyväksymät standardit. Työssä vertaillaan kahta standardia, joissa kuvaillaan menetelmiä äänitehotason määrittelyyn taajuuskaistoittain riittävällä tarkkuudella. Nämä standardit ovat ISO 3744 ja ISO 9614-2.</p> <p>ISO 3744 määrittelee useamman menetelmän, joilla voidaan määrittää äänitehotasot mittaamalla äänenpaine diskreeteissä mittauspisteissä. ISO 9614-2 kuvailemassa menetelmässä äänitehotasot määritellään ääni-intensiteetin avulla. Ääni-intensiteetti approksimoidaan skannaamalla äänilähteen pintaa ääni-intensiteettisondilla. Vertailemalla näiden standardien kriteereitä mm. mittaustilanteiden ja mittalaitteiden suhteen, päädytään tässä työssä suosimaan standardia ISO 9614-2.</p> <p>Lisäksi työssä esitellään muutamia meluntorjuntamenetelmiä, vaikka taajuusmuuttajien tuottamat äänenpainetasot normaalioloissa harvoin ylittävät vaarallisen tason. Tästä syystä tämäkin työ painottaa enemmän luotettavan mittaustulosten keräämistä kuin meluntorjuntaa. Tämä työ ei vielä itsessään riitä perusteena toteuttaa tiettyä meluntorjuntamenetelmää taajuusmuuttajiin, vaan kyseiset menetelmät vaativat lisätutkimusta. Tässä työssä kuitenkin tutkitaan myös käytännössä yhden absorbtiomateriaalin vaikutusta äänilähteen tuottamaan spektriin.</p> <p>Työtä varten tehtiin useita äänimittauksia ilma- ja nestejäähdytteisille taajuusmuuttajille eri operointipisteissä. Testien tulosten perusteella voidaan huomata, että mittaustapa tuottaa koherenttia ja luotettavaa dataa, vaikka mittausten tekijöillä ei olisikaan kokemusta kyseisestä standardista. Työn lopputuloksena voidaan määritellä toimenpiteet, joilla kaapitettujen taajuusmuuttajien äänimittaukset voidaan tulevaisuudessa toteuttaa nopeammin ja pienemmällä vaivalla.</p>			
Asiasanat:	taajuusmuuttaja, melu, melunmittaus, meluntorjunta, ääni-intensiteetti, ääni-intensiteettimittaus, ääniteho, äänitehotaso, äänenpaine, A-painotus		
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Espoo, December 12, 2018

Antti Hepo-oja

Abbreviations and acronyms

AC	Alternating current
B&K	Brüel & Kjær
BNC	Bayonet Neill–Concelman connector
BPF	Blade pass frequency
DC	Direct current
EUT	Equipment under test
IEC	International Electrotechnical Commission
IIR	Infinite impulse response
IP Code	International Protection Marking
ISO	International Organization for Standardization
LRAM	Locally resonant acoustic metamaterial
MPP	Micro perforated plate
PI index	Pressure-intensity index
PWM	Pulse-width modulation
RMS	Root mean square
SPL	Sound pressure level
SWL	Sound power level
VTT	Technical Research Centre of Finland

Symbols

ω	Angular frequency
f	Frequency
λ	Wavelength

f_0	Resonance frequency
ρ	Density of the medium
P	Sound power
L_W	Sound power level
γ	Adiabatic index
γP_0	Bulk modulus
c	Speed of sound
$\partial V/V_0$	Volumetric strain
p	Sound pressure
L_p	Sound pressure level
u	Particle velocity
I	Sound intensity
E	Young's modulus
ν	Poisson's ratio
G	Shear modulus
c_l	Longitudinal waveform velocity
z	Wave impedance
S	Measurement surface

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Chapter 1

Introduction

The aim of this thesis is to find a method for noise measurements of cabinet-built frequency converters that is easy to repeat and generates reliable noise data. Subsequently, the validity of the method is established through test cases. This thesis describes methods of noise level measurements which are defined in two standards that are widely adopted in various industries. These standards are ISO 3744 and ISO 9614-2. We compare the viability of the described methods in the scenario where cabinet-built frequency drives are measured. Out of these two standards, one method is picked based on the advantages and the disadvantages of both standards. The method is then tested with several test cases, and the results are analyzed, compared, and discussed. In addition, relevant acoustic phenomena related to concepts in noise measurements, such as sound propagation and A-weighting, are discussed.

As a result of this thesis, a comprehensive noise test code is produced. The test code can be followed to obtain the total A-weighted sound power level and the total A-weighted sound pressure level at a distance of one meter. In addition, the method can be used to produce more detailed data in either octave bands or one-third octave bands. This data can be analyzed to recognize those frequency bands that output the highest sound levels. This data gives indication about the nature of the noise sources within a frequency converter, and gives a realistic view on the total noise output of frequency converters.

When the frequency bands with highest noise levels are recognized, the information can be used to attenuate these levels. Several noise control methods are discussed that can be used to this task. These noise control methods include resonators, metamaterials, and controlling the sound waves in order to superimpose signals destructively. Although these noise control methods are not extensively tested in practise in this thesis, some interesting

and favorable methods are found.

In addition to generating an applicable noise test code for cabinet-built frequency converters, this thesis aims to suggest further steps in investment and research as the test cases included in this thesis are hardly enough to give a complete view on the noise output of cabinet-built frequency drives.

In chapter 2, we discuss frequency converters as noise sources and describe some general phenomena in acoustics. In chapter 3, an insight is given on measurement instrumentation, noise measurement methods, and methods of noise control. Chapter 4 describes the measurement setup (including measurement instrumentation and measurement environment in use) and test cases. In chapter 5, the results for the test cases are presented and discussed. Finally, evaluation of the validity of the chosen method and suggestions on further steps are given in chapter 6.

Chapter 2

Background

2.1 Frequency converters

A frequency converter is a device that is used to convert the frequency and voltage of the AC supply network into another frequency and voltage which are subsequently fed into another electrical circuit. Frequency converters can be used to control AC motors, i.e. load, at specified speeds by converting alternating current from the supply grid (at 50 Hz or 60 Hz) to direct current and back to AC at a specified frequency and voltage according to given parameters. Some applications include cranes, conveyors, centrifuges, elevators, and pumps. Vice versa, the frequency converter can be used to transfer energy from the motor to the supply network. The parameters are specified by the requirements of the process that the frequency converter is a part of. For example, elevators should operate as smoothly as possible, which requires continuous control over motor speed. Without a frequency converter, the motor could be driven only at the frequency of the supply grid.

In this thesis, the word *drive* refers to a high-power cabinet-installed frequency converter. These frequency converters either use dry clean air from its surroundings for cooling or coolant liquid to cool down the circulated air within the cabinet.

Frequency converters are used in a variety of industrial environments e.g. marine and metal industry. They are also used in harsher environments such as in mining industry and waste water treatment. The operating environment of the frequency converter sets requirements on the design of the drive. For example, harsh environments may require enclosed cabinets or better filtering around the air inlets, both of which inevitably affects the sound that the device generates [1].

A cabinet-built frequency converter consists of multiple cabinets that in-

clude the supply unit and one inverter unit per controlled motor. The supply unit, which converts AC to DC, is connected to the supply network. The inverter unit, which converts DC to AC to control the motor, is subsequently connected to the supply unit via DC link [2]. These two units form the basis of how drives function, but additional units may be included depending on the layout of the drive. For example, the auxiliary control unit contains I/O for external devices and brake units are used to dissipate extra energy from the drive circuit by braking a high-inertia motor when a certain voltage limit is exceeded [3, 4]. Other non-standard options may also reduce the overall noise level. For example, sine filters can be used to create near sinusoidal output voltage in order to improve the quality of the output current that is fed to the motor [5] and to reduce the motor noise level [1]. However, motor noise is not further discussed in this thesis as only noise from the frequency converter is investigated.

Liquid-cooled frequency converters rely on a liquid coolant to cool down the air that absorbs the heat losses from the components that heat up. As a result, no air flow between the cabinet and the operating environment is needed to generate air flow to transfer the heat. As the cabinet is fully enclosed, the device is also quieter and more suited for harsh operating environments [6]. The effect of a liquid coolant as a choice of cooling on the emitted noise level is further discussed in section 2.7.3.

Although the main concept is similar in each device, it is difficult to specify the device more strictly as the order of the cabinets in the device and the layout within the units may vary [3]. Additionally, as the structure and consequently the noise level depend on the cooling type of the drive, it needs to be specified whether a liquid cooled drive is under consideration. In this thesis, the drives under examination have no additional equipment or options installed unless otherwise specified.

2.2 Generation of sound and noise

Sound can be described as the fluctuation of density in a fluid medium in a time unit. In other words, sound is a variation in local instantaneous pressure that exists in a medium as a propagating waveform. The faster the rate of this variation is, the higher the sound generated as the disturbance in the medium is balanced. [7] This variation can be recorded or analyzed [8].

Perceived sound involves a variety of mechanisms to translate physical sound waves into an auditory event. These mechanisms are discussed in section 2.5. Bone conducted vibrations can also contribute to an auditory event at low frequencies or when the source is very near to the observer

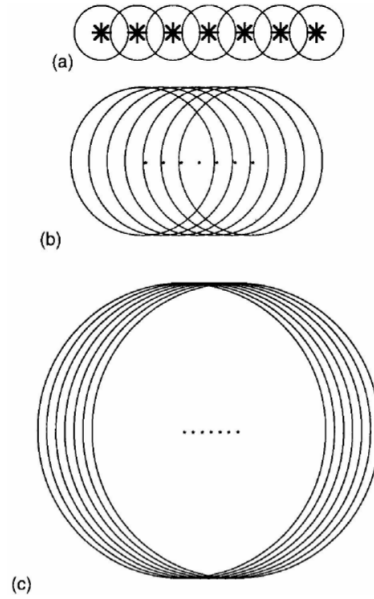


Figure 2.1: The behavior of wavefronts in a), b) near-field and c) far-field [7].

[8]. However, bone conducted sound is perceived quieter than air conducted sound [8], and for this reason the phenomenon is not relevant to this thesis. Noise is generally undesired and unnecessary sound generated within a device or equipment. Sound usually propagates as longitudinal waves, but it can also propagate as traversal acoustic frequency waves in solids [8, 9]. Hearing mechanisms and propagation of sound are discussed more extensively in sections 2.5 and 2.4.

A sound source is a system that produces the sound, usually a vibrating mechanical structure. Sound sources can be categorized by their properties, for example with the method of sound generation and the directivity of the radiation pattern. The simplest case of a sound source is a point source. As its name suggests, a point source is a sound source with negligible dimensions, for example in cases where the observation distance to the source is increased long enough. [9]

When the distance to the sound array is increased as the wavefronts propagate, the radiation pattern gain characteristics similar to spherical sources [7, 9], and the observer is considered to be in far-field [10]. This behavior is shown in the figure 2.1. An array of monopoles can form a multipole, such as a speaker with tweeter, midrange, and woofer. Multipoles give more information about the directivity of the radiation pattern whereas monopoles carry the information about the overall sound level [11].

Even if the sound propagates in a spherical or hemispherical manner, the

radiation pattern can be biased into any direction [7, 9]. Consider a computer case, or any other air-cooled device: the noise propagation is biased into the direction normal to the axis of the cooling fan. In the absence of the case, the sound field is biased radially perpendicular to the axis of the fan [7].

A sound can be impulsive, caused by a transient excitation such as the hand-clap or it can be continuous like the humming of the air ventilation system [9]. Obviously, a noise source can radiate a combination of these static and transient sounds. The characteristic frequency spectrum of the sound source contains valuable information about the nature of the sound. If the spectrum contains peaks at higher frequencies within the most sensitive hearing range, the sound may be considered more irritating than noise with a smooth spectrum. Broadband noise, in contrast to narrowband noise, radiates a frequency spectrum with considerable amount of energy in a wide range of frequencies. Measurement of frequency spectra are of particular interest in this thesis [10].

Although we do not have the means to affect sound generation directly given the scope of this thesis, radiation from thin plates and fans are discussed more extensively to determine the primary noise sources within frequency converters. It would be convenient to examine frequency converters as point sources. The related sound pressure and radiated sound power for a point source (or when observing a source in a far-field) are given by the following equation [9]:

$$p(r) = \frac{j\omega\rho}{4\pi r} Q e^{-jkr} \quad (2.1)$$

$$P = \frac{\omega^2 \rho Q^2}{8\pi c} \quad (2.2)$$

where ω is the angular frequency, ρ is the density of the medium, r is the observation distance, c is the speed of sound and the amplitude is given by [9]:

$$Q = 4\pi a^2 \nu(a) \quad (2.3)$$

where a is radius of the noise source that is modeled with a sphere and $\nu(a)$ is the radial velocity amplitude on the surface of the point source [9]. However, we very rarely examine drives in far-field, but instead the observer is usually relatively near to the device. Drives do exhibit a property of directivity in near-field in contrast to far-field. For this reason, we will examine drives in near-field as an array of uniformly radiating surface segments.

A mechanical system can have one or more resonant frequencies where parts of the system vibrate at the same frequency. The vibration of a mechanical structure, which subsequently produces a sound, can be boosted by

hitting near one of the resonant frequencies of the vibrating system. Resonance is often undesired outside of musical instruments as it can amplify the noise level or cause malfunctions in the system. Vibration is naturally attenuated through energy losses such as heat or electrical vibrations. Excessive vibrations can be attenuated through correct damping. [8]

However, resonators can also be used in noise control: Helmholtz resonators, for example, are widely adopted to this task [8, 9]. Use of Helmholtz resonators in noise control is discussed in section 3.3.2.

If two or more sound sources are related in phase and frequency, they are considered to be *coherent*. The phase and coherency of the sound sources affects whether the sound sources add constructively or destructively when they are superimposed. Exploiting this property of sound signals, it can be harnessed for noise control by destructively adding a noise signal to an undesired noise signal [8]. It is assumed that in the case of this thesis, the equipment under test has several different noise sources, each with different phase and frequency range. As a result, we are observing an incoherent system of sound sources.

Studying and mitigating the noise output is not a trivial topic. As it is well known, noise can cause hearing degradation or even hearing loss. It is also known to increase stress levels and decrease the quality of life [7]. Even if the sound power level or sound pressure level is below safe limits, it can be perceived irritating. The importance and rise of interest in this matter can be seen in the consumer market. For example, the home appliances often advertise their silent performance as it has become a valuable feature for consumers. Efficiency does not necessary correlate with the loudness, and reducing structure-born sound reduces malfunctioning of the device as vibrating structures are attenuated. Some methods for mitigating the typical contributors to noise in a drive are discussed in next chapter, section 3.3.

2.3 Related acoustic quantities

In this section, a description of few acoustic quantities are given. In this thesis, we are mainly interested in sound pressure, sound intensity, and sound power and their relative logarithmic values. These quantities are used in the measurements as described in sections 3.2.1 and 3.2.2.

2.3.1 Sound pressure and sound pressure level

As stated in the standards ISO 3744 and ISO 9614-2, sound pressure is a scalar quantity that describes the difference between the instantaneous

pressure and the static pressure. Static pressure is the pressure that takes place in the absence of sound [8]. Sound pressure is defined as a RMS value as the average of the amplitude is zero [10]. The unit of sound pressure is the Pascal, denoted by $[Pa]$. The range of human hearing as a function of sound pressure is $20 \cdot 10^{-6} - 50 Pa$ [8].

However, human hearing does not perceive change in sound pressure linearly but rather in a logarithmic manner. To better represent the characteristics of the human hearing, a concept of sound pressure level has been defined [9]. Such logarithmic scales have been defined for many other acoustic concepts, too. As noise signals are not usually coherent, the logarithmic scale is multiplied by 10. For coherent signals, the multiplier is 20. [10]

To begin to define the sound pressure level, we first need to obtain the acoustic pressure. In sound waves, the local temperature varies as a function of the pressure and the density of the medium. This temperature difference gives rise to a heat flow to balance the temperature. However, the sound wave can be assumed to propagate adiabatically in high frequencies as the heat flow does not have enough time to occur. [7]

The pressure in an adiabatic process can be determined with the following relation [7]:

$$P = \alpha \rho^\gamma \quad (2.4)$$

where α is a constant, ρ is the density of the medium, and γ is the adiabatic index. The change of pressure in relation to the change of density in equilibrium can be derived from the state equation for ideal gases ($P/\rho = RT$) where R is a coefficient related to the particular gas [7]:

$$\left(\frac{\partial P}{\partial \rho} \right)_0 = \gamma P_0 / \rho_0 = \gamma R T_0 \quad (2.5)$$

where P_0 , ρ_0 , and T_0 denote pressure, density, and temperature in equilibrium, respectively. The constant γP_0 is the bulk modulus which describes the compressibility of the gas. The propagation speed of the sound wave is related to the bulk modulus by [9]:

$$c^2 = \frac{\gamma P_0}{\rho} \quad (2.6)$$

The change of density in relation to change of volume can be determined with following relation [7]:

$$\frac{\partial \rho}{\partial V} = -\frac{\rho_0}{V_0} \quad (2.7)$$

where V_0 denotes the volume in equilibrium. From equations (2.5) and (2.7), acoustic pressure (total pressure without static pressure) can be derived with the following equation [7]:

$$p = (\partial P / \partial \rho)_0 \delta \rho = -(\gamma R \rho_0 T_0) (\delta V / V_0) \quad (2.8)$$

where $\delta V / V_0$ describes the volumetric strain, discussed in section 2.4. Sound pressure level can be calculated from the acoustic pressure p and the reference value $p_0 = 2 * 10^{-5} \text{ Pa}$ with the following equation [9]:

$$L_p = 10 \log_{10} \left(\frac{p}{p_0} \right)^2 \quad (2.9)$$

The reference value represents the human hearing threshold at range from 1 kHz to 3 kHz [9]. If the sound consists of two incoherent signals, the total sound pressure level for the signals is [12]:

$$L_p = 10 \log_{10} (10^{L_{p1}/10} + 10^{L_{p2}/10}) \quad (2.10)$$

2.3.2 Sound energy and sound energy level

Sound energy is comprised of the kinetic energy and the potential energy carried by the sound wave, former being proportional to the square of particle velocity and latter to the square of sound pressure. The unit of sound energy is the Joule [J]. The kinetic energy E_k and potential energy E_p can be obtained with the following second order equations when the heat flow within the medium is negligible: [7]

$$E_k = \frac{1}{2} \rho_0 u^2 \quad (2.11)$$

$$E_p = p^2 / 2 \rho_0 c^2 = p^2 / 2 \gamma P_0 \quad (2.12)$$

Sound energy is then defined as follows [7]:

$$E = E_k + E_p = \frac{1}{2} \rho_0 u^2 + \frac{1}{2} p^2 / \rho_0 c^2 \quad (2.13)$$

2.3.3 Sound intensity

Another useful concept in terms of noise control is acoustic intensity that describes the energy carried by the sound wave through a hypothetical surface. The main advantage is that measuring the sound intensity allows us to

examine sound sources in field conditions. Sound intensity is proportional to the particle velocity and the sound pressure. [13]

To begin deriving the relation of the sound intensity to the sound pressure, we take advantage of the instantaneous sound intensity in the absence of heat flow and in the presence of volumetric strain only [7]:

$$\mathbf{I} = p\mathbf{u} \quad (2.14)$$

where \mathbf{u} is the vector quantity of acoustic particle velocity in a wave. As a result, the sound intensity is also a vector quantity. The unit of sound intensity is presented as the ratio of sound power per surface, denoted by $[W/m^2]$ [13]. The rate of change of energy density can be derived from the equations (2.13) and (2.14) [7]:

$$\delta E/\delta t = -[\delta(pu_x)/\delta x + \delta(pu_y)/\delta y] = \frac{\delta I_x}{\delta x} + \frac{\delta I_y}{\delta y} \quad (2.15)$$

where u_x and u_y are the magnitudes of particle velocity in their respective axis. In far-field, sound intensity can also be approximated with the following equation [9]:

$$I = \frac{p^2}{\rho c} \quad (2.16)$$

The sound intensity level is given by the following equation [13]:

$$L_I = 10 \lg \left(\frac{I}{I_0} \right) \quad (2.17)$$

Where the reference value I_0 is $10^{-12} W/m^2$. The sound intensity level is close to the sound pressure level in far-field and free-field [9].

2.3.4 Sound power and sound power level

Sound power represents the sound energy generated by the sound source in a timeframe [7]. As stated in the standard ISO 9614, sound power is given by integrating sound intensity over a hypothetical surface enclosing the noise source. The unit of sound power is the Watt, denoted by $[W]$. Essentially, sound power is the total power radiated by the source.

Sound power can also be acquired with the following equations from the sound pressure [10]:

$$P = \frac{1}{\rho c} \int p^2(s) dS \quad (2.18)$$

$$P = \frac{4\pi r^2}{\rho c} \frac{1}{N} \sum p_i^2 \quad (2.19)$$

The former equation is a mathematical representation that cannot be used for measurements in practice as continuous microphone arrays are impossible to implement. The latter equation can be used for measurements as it allows discrete measurement positions.

Sound power level can be acquired from sound power with the following equation [13]:

$$L_W = 10 \lg \left(\frac{P}{P_0} \right) \quad (2.20)$$

where P_0 is the with value 10^{-12} W .

The efficiency of the sound source describes how much of the power of the source is transformed into sound power $\eta = P_a/P$ [8]. In noise control, the sound power level is considered a useful quantity as it is not affected by the properties (geometry or surface materials) of the room unlike the sound pressure level [13, 14].

2.4 Sound propagation

Sound waves carry kinetic energy, momentum, and potential energy through density fluctuations as they propagate. Some of the energy is dissipated through viscous mechanisms or it can be reflected or transmitted at the interface of two media as discussed in next section. [7] In this section, we examine how sound propagates in different media.

2.4.1 In fluids

Sound waves propagate in a longitudinal waveform in fluids. Fluids are media that lack resistance to shear stress, i.e. liquids and gases [7, 9]. The sound waves transmit energy and momentum at the speed of the wave. However, particles are transmitted at a negligible rate. The sound wave continues to propagate until the energy associated to it is finally dissipated through heat and viscous mechanisms. [7] Air absorption is not accounted for in this thesis as the measurements are conducted in near-field.

Fluids, such as air, are fairly elastic. Elasticity, which describes the ability of the fluid to resist volumetric strain, allows for the displacement of fluid particles, causing local pressure maxima and minima within the fluid. Fluids also have a property of fluid inertia that resists elastic reactions caused by the volumetric strain. [7]

A layer of molecules in fluid resists the movement of adjacent layers with different momentum in order to maintain mean velocity within the medium. The effect that maintains the mean velocity is called viscosity. Viscosity is also present in fluids with tangential velocity difference to the interface with a solid media, and some of the energy may be transmitted to the atoms in the surface of the solid. [7, 9] In gases, viscosity increases as the temperature increases. Viscosity dissipates sound energy naturally as viscous stresses are not conservative [7]. Therefore, it can be exploited in noise control as discussed in next chapter.

Other dissipative mechanisms in gases are concepts coined as thermal diffusion and molecular relaxation [7]. Thermal diffusion occurs when there is a difference of density, pressure or temperature in two regions within the fluid. Heat flow occurs to even out this difference and ultimately some of the energy associated to the sound wave is lost. Molecular relaxation is the phenomenon where at a sudden compression of gas, the molecules receive translational kinetic energy, some of which is transformed into rotational and vibrational energy of the molecules. [7, 9] This phenomenon is irreversible and some of the energy is lost if the compression is reversed sufficiently slowly. However, as is the case with viscous stresses, these mechanisms are not efficient at attenuating sound levels in dry air. [7]

The speed of sound depends on the bulk modulus of the fluid which gives a value to the elasticity of the fluid [7, 9, 15]:

$$c = \sqrt{\gamma RT_0} \quad (2.21)$$

The wave equation in three dimensions is given by the following equation [9]:

$$\frac{\partial^2 p}{\partial t^2} = c^2 \nabla^2 p \quad (2.22)$$

2.4.2 In solids

In solid media, such as thin plates, the sound propagates in three waveforms: longitudinal waves, shearing waves, and bending waves [9].

Shearing and bending waves are transverse waves that oscillate with motion perpendicular to the propagation direction. They are able to propagate in solids as they have resistance to shear stresses, which balances disturbances. A shearing wave is purely a transverse wave with motion perpendicular to propagation, whereas bending waves contain also motion in the direction of propagation. [7]

A vibrating string is an example of transverse waves in one dimension, whereas a thin membrane can support transverse waves in two dimensions. Out of the three transverse waveforms, bending waves are the most significant contributor to structural-borne sound [9]. Solid bars can also support a third type of a transverse wave called torsional waves which are of little importance in this thesis [7].

There are three properties that affect the modal frequencies of a vibrating membrane: air loading, bending stiffness, and shearing stiffness of the membrane. Thin plates are essentially membranes that have a prominent property of stiffness. What comes to the boundary conditions, thin plates can be either hinged, clamped, or free. [9] For example, the metal plated cabinet of the frequency converter can be modeled as a collection of clamped thin plates. Cabinet plates are sturdy enough, i.e. they have a high stiffness, so that they can be considered as thin plates.

The velocity of pure longitudinal waves in a thin plate can be obtained with the following equation [9]:

$$c_l = \sqrt{\frac{E}{\rho(1 - \nu^2)}} \quad (2.23)$$

Where E is Young's modulus, ρ is the density of the solid medium, and ν is Poisson's ratio. If the dimensions of the plate are smaller than the wavelengths associated with the sound wave, the vibrations propagate as *quasi-longitudinal waves* instead of pure longitudinal waves [9].

The velocity for a quasi-longitudinal wave in a thin plate can be obtained with the following equation [9]:

$$c'_l = \sqrt{\frac{E(1 - \nu)}{\rho(1 + \nu)(1 - 2\nu)}} \quad (2.24)$$

Shearing waves cause deformations in the solid, and the solid tend to resist this change, attenuating the motion of the wave. Shearing waves are slower than the longitudinal waves as the shear modulus G is less than Young's modulus. [9]

Shearing waves propagate at the velocity given by the following equation [9]:

$$c_t = \sqrt{\frac{G}{\rho}} \quad (2.25)$$

The velocity of the most significant form of transverse waves, bending waves, in thin plates depends on the frequency of the waveform [9]:

$$c_b(f) = \frac{\omega}{k} = \sqrt{1.8fhc_l} \quad (2.26)$$

Where h is the thickness of the plate, and c_l is the velocity of the longitudinal waveform. The frequency f can be approximated with the following equation [9]:

$$f = \frac{\omega}{2\pi} = 0.0459hc_l k^2 \quad (2.27)$$

Where k is the respective value for the normal modes of vibrating motion.

In case of transverse waves, as propagation speed increases when frequency increases, the mechanical structure can have additional inharmonic modal frequencies. A thin plate is a two-dimensional membrane with prominent stiffness that has multiple resonant modes of different geometries. [8]

2.4.3 Wave interaction

As sound waves propagate through medium, they can *interfere* with each other through superposition of the wavefronts. If the sound sources are coherent, the resulting pattern exhibits local interference maxima and minima. In room acoustics, interference can create standing waves with a large amplitude at the resonant frequencies of the room. Different frequencies create different interference patterns i.e. acoustic modes. Even if room acoustics are not particularly interesting considering the scope of this thesis, these acoustic modes can also occur in enclosures. [7]

Naturally, sound waves will eventually meet obstacles e.g. solid objects or variations in the density of the medium as they propagate. For example, the sound wave reflects almost completely as it propagates from air to much denser water [9]. To simplify the case, it is assumed that the object is large in contrast to the wavelength of the incident sound wave. In such a case, sound waves obey the Snell's law of reflection. When the sound wave comes upon an obstacle of porous material, a significant amount of its energy will be dissipated into heat. [7] This is discussed more extensively in next chapter.

If the obstacle has a non-smooth surface with discontinuities or it is smaller than the wavelength of the sound wave, the incident sound wave can be scattered in different directions. As a result, the sound energy of the wave is also scattered. If the sound wave hits the resonant frequency of the object, scattering is more efficient. [7]

The propagating sound waves can be refracted at a boundary of two media, meaning that the speed and the direction of the sound wave will change [7, 9]. If the change in properties of the media is gradual, the reflection

of sound is less significant. This phenomenon can be observed when an airplane passes the observer at clear night; the temperature gradient from faster cooling ground to warm air causes the sound waves to bend downwards [16]. Additionally, the wind conditions can cause sound waves to bend [7, 10].

When the change in medium is not gradual, for example at a discontinuity between a fluid and a solid medium, most of the energy from the sound wave reflect and some of it is transmitted into the medium. When the wave is transmitted, it can propagate further as acoustic frequency waves or the acoustic energy associated to it can be completely transformed into energy losses depending on the properties of the material. [8] If the sound wave has enough energy, it can reach the opposing surface. This surface will then act as another sound source as the surface radiates a portion of the transmitted energy as sound waves [8, 9].

When a sound wave transmits from a fluid to a solid, part of the transmitted sound wave transforms into transverse waves. In special cases, such as normal incident sound waves on infinite extended thin plates, the transmitted wave can continue propagating solely as a longitudinal wave. The transmission and reflection coefficients, respectively, are given by the following equations [9]:

$$\frac{B}{A} = \frac{z_2 - z_1}{z_2 + z_1} \quad (2.28)$$

$$\frac{C}{A} = \frac{2z_2}{z_2 + z_1} \quad (2.29)$$

where A is the amplitude of the incident sound wave, B is the amplitude of the reflected wave, C is the amplitude of the transmitted wave, and z_1 and z_2 are the wave impedances of respective media.

The phase of the reflected wave depends on the wave impedances. If the wave impedance of the medium where the incident wave propagates is smaller than the medium of the obstacle, the reflected wave will be in phase. Otherwise the reflected wave is in opposite phase with the incident wave. [9] As it can be seen from the equation (2.28), the reflection coefficient increases as the difference between wave impedances increase. The angle of incidence determines the amplitude of the transmitted and reflected waves [9].

2.4.4 Absorption

As the sound wave propagates through a medium, some of the energy is dissipated e.g. due to viscous forces and thermal losses. Attenuation coefficient describes how quickly the sound wave attenuates as it propagates in

the medium. For air at 50% relative humidity, the attenuation coefficients are given by [9]:

$$\alpha \approx 4 \cdot 10^{-7} f, \quad 100 \text{ Hz} < f < 1 \text{ kHz} \quad (2.30)$$

$$\alpha \approx 1 \cdot 10^{-10} f^2, \quad 2 \text{ kHz} < f < 100 \text{ kHz} \quad (2.31)$$

At small distances air absorption can be neglected. To cut off frequencies around 5 kHz, the sound wave needs to travel approximately 100 meters [10]. In this thesis such distances are irrelevant as user cases assume that the observer is directly next to the device or in the same room.

Acoustics address the problem of how the environment surrounding the sound source affects the sound propagation. In enclosed spaces, sound levels can be affected by controlling the reflections and reverberation properties of the space. The sound event consists of the direct sound and secondary sounds caused by reflections. Reverberation is caused by the secondary sound waves that are reflected from the walls. [8] For impulsive sounds, this can be heard as the decay of sound after the direct sound. The absorption coefficient describes the ability of the corresponding material to transmit and absorb the energy of the incident sound wave. [8] Although room acoustics are out of scope for this thesis, proper acoustic design of the room is not a trivial matter as rooms with poor acoustic properties can actually amplify the undesired noise.

If the wavelength of the sound wave is long relative to the discontinuity that the wavefront encounters, some energy associated to the sound wave can be diffracted around the boundary as explained by Huygens' principle. According to the principle, the wavefront of the sound wave is a source of secondary sound waves or wavelets. For this phenomenon, we are able to hear sounds behind obstacles, but it also makes noise control more difficult. [7, 8]

2.5 Human hearing

Typical human auditory system can perceive frequencies ranging from 20 Hz to 20 kHz, translating broadband sound into frequency bands. Hearing is most sensitive from 100 Hz to 10 kHz. Consequently, the resolution of human hearing is better at certain frequency bands.[9] Analysis of signals at frequency bands gives us valuable information of how the sound source is perceived.

In human auditory system, the vibrations of the air propagating into the ear canal are transferred into the inner ear through a collaboration of ear

drums and ossicles [9]. Ossicles are used for impedance matching to transmit incident sound waves into the medium in the inner ear. This mechanical vibration is translated into neural activity through the collaboration of the basilar membrane and inner hair cells. The perceived sound is regulated by the non-linear activity of the cochlea in the inner ear. [8] In theory, the middle ear can slightly protect the inner ear from loud noise at low frequencies by controlling the conductivity of the ossicles, but the effect suffers from slow response time to impulsive sound (10-100 ms) [10].

Loudness is a psychoacoustic quantity which describes how loud the sound is perceived. The unit of loudness is *sone*. The perceived loudness depends proportionally on the length and the type of the sound signal; broadband white noise is considered louder than a tone. [8] In addition, loudness depends on the type of the sound field (free-field or diffuse-field) [17].

Relatively high amplitude sine signals, on the other hand, can be considered more irritating and thus more undesirable than white noise as they generate noticeable peaks in the frequency spectrum. As expected, loudness also depends on the frequency of the sound; low frequencies are not perceived as loud as higher frequencies even if they share the absolute sound pressure level. Dangerous sound levels can go by unnoticed if the frequency of the sound is low enough. [8]

Loudness level describes the sound pressure levels for the complete human hearing range where the sounds are considered equally loud to a reference sound. From this definition, psychoacoustic tests have been conducted to obtain equal-loudness curves using a reference tone of 1 kHz at a certain sound power level. The unit of loudness level is phon. [9] Psychoacoustic tests have shown that loudness does not increase linearly as the sound pressure level increases [8]. This behavior is modeled with equal loudness contours as shown in figure 2.2.

The hearing threshold is the sound pressure level where an auditory event is generated. The value of the threshold is defined as 0 dB. The pain threshold is at around 120 dB, whereas the human hearing has the lowest sensitivity around 1-3 kHz. [9] The threshold for immediate hearing degradation or even hearing loss is at around 130 dB. 85 dB is considered to be the limit for harmful sound pressure level when continuously exposed to such noise. [8]

Roughness is also a psychoacoustic quantity relatively important to this thesis caused by the amplitude modulations of the signal [18]. In presence of two signals, depending on the difference of the frequency of the signals, the interference of the two superimposed signals can cause beating of the sound. With slightly higher difference in frequency, the resulting signal gains prominent roughness. [8] Rough or beating sounds can be considered irritable and are thus undesirable.

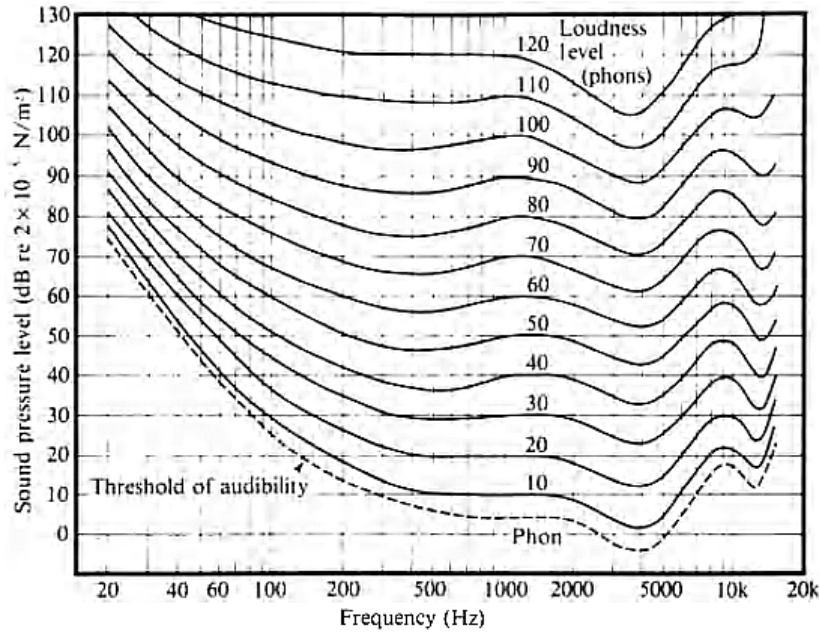


Figure 2.2: Equal loudness contours, sound pressure level as a function of loudness level [9].

The effect of the non-linear sensitivity in the human auditory system when measuring the sound levels is represented with the A-weighting of the sound spectrum. A-weighting curves are used to emphasize mid-frequencies in a frequency spectrum to correlate with the frequency sensitive property of the human hearing. A-weighting values are predetermined with psychoacoustic tests, presented in the standard IEC 61672.

2.6 Spectral masking

A magnitude spectrum, representing the amplitudes of energy associated to different frequencies within a sound signal, is a useful tool to find abnormalities or to analyze the sound source. For example, we can see whether the sound source emits a discrete, high-frequency tone, or if it is a broadband sound source. Human auditory system is able to decompose an incoming sound signal into a spectrum of frequency bands with certain center frequencies to distinguish different frequencies. However, this ability is held back by the frequency resolution of the ear.[8]

Masking is the phenomenon where certain frequencies in the spectrum cannot be perceived in the presence of another sound. The cause of this

effect lies in the linear and non-linear features of human hearing. Masking threshold is the value of the sound pressure level at a certain frequency that is required for the original sound to be audible with the masker sound present. The loudness of the original signal decreases as the level of the masker is increased. Masking threshold curve is used to determine the masking threshold for different frequencies. Even if the masked sound cannot be heard, the superposition of the masker and the sound can cause undesired and unnecessary noise such as beating. [8]

Masking can be achieved with several techniques. One of these techniques includes using broadband noise. The spectral density of the broadband noise determines the masking threshold. Using broadband noise as a masker sound is a viable option when it is required to completely mask out a certain sound. Temporal masking is the concept of masking sounds in time, for example playing a louder sound before a sound with a low amplitude masks the latter sound. [8]

Narrow-band noise can be used to mask certain peaks in the sound spectrum. The center frequency of the narrow-band noise is used to control which peaks are masked. Additionally, increasing the sound pressure level of the masker increases the range of masked frequencies above the center frequency. If there are tones near to the frequency of the masker, beating effect can be observed. [8, 18] More complex masking threshold curves can be achieved by using complex harmonic tones where different harmonics contribute to different parts of the masking pattern [8].

In this thesis, we are interested in an equipment that produces continuous, broadband noise, thus temporal masking is not an efficient noise control technique. Furthermore, increasing the noise level at certain frequency bands is not an ideal solution if the overall noise level is examined.

2.7 Main contributors to drive noise

The operating conditions of the drive indirectly contribute to the noise output of a generic drive. The difference between the no-load case and maximum capacity is obvious when considering noise level. As the output current level is increased, the drive will generate more heat. As a result, the air flow needs to be increased to maintain appropriate temperature within the device. This leads to increased noise through increased fan rotation speed.

In this section, the main contributors to the noise radiated by a generic drive are discussed. In this case, noise is determined to be excessive, undesired sound generated by the sub-components of the device that is deemed irritating or harmful. The measures that can be taken to suppress the noise

caused by the sub-components are discussed in the next chapter.

2.7.1 Fans

A fan is a relatively small structure of several rigid blades that rotate around the axis. The main contributors to the noise output of a typical frequency converter are the PWM controlled DC inlet fans used for cooling the modules within the unit cabinets. In addition, outlet fans are located on the top of the cabinet to maintain the air flow through the drive, but these fans are not considered to contribute as much to the noise level.

The aerodynamic force that is imposed on the blades of the fan has two time-steady components: lift component along the axis of the fan, and drag perpendicular to the axis. The fans radiate noise through vortex shedding, perpendicular to the axis when the blades are rotating through the air mass. Vortex shedding is dependent on the blade pass frequency (BPF), determined by the angular frequency and the number of blades of the fan. [7, 19] The fan radiates steady-loading noise linked to the BPF [20].

Fans are used to give rise to the air flow to cool down heated components and subsequently pushing out the heated air from the modules. The more powerful the equipment, the more efficient cooling is generally required. This typically means increasing the air flow in order to cool down the components, but also liquid cooling can be used to cool down the air coming from the inflow. If air cooling is used, BPFs of the fans are increased, and consequently, the steady-load noise output is also increased. Additionally, it is often required that the fan diameters are smaller in order to create more compact devices, which in turn means that the revolution speed of the fan must be increased to maintain the flow rate. Unfortunately, increasing the fan diameter is not an option in this thesis either, which would naturally decrease the rotation speed required for a certain flow rate.

Any kind of pressure drop should be avoided when it comes to the operation of fans as it reduces the flow rate. Pressure drop is a decrease in static pressure difference between the device and the environment [3]. Pressure drop can be observed for example if air filtering is used; the more dust the filters gather, the more significant the pressure drop. In addition, the distance of the fan from any obstacle affects the flow and the efficiency of the fan. If the obstacle is too near to the fan, pressure drop can be observed and the fan efficiency suffers. Furthermore, the air flow from the inlet to the outlet fan may turn unstable if the flow is disturbed, giving rise to turbulence [19].

The efficiency of a fan depends on the diameter of the fan and the shape of the blades. There are different categories of fans based on the shape of

their blades. Each type has their advantages and disadvantages. The benefit of axial fans is the fact that motor overloading is not possible. Centrifugal fans with forward-curving blades typically have low noise levels and high pressure rise. However, motor overloading is possible. Centrifugal fans with backward-curving blades have higher noise level than such fans with forward-curving blades, but they also have better efficiency and motor overloading is less likely. The flow rate of a system also depends on how the fans have been installed. If the fans are installed in parallel, it can be observed that the flow rate is doubled. If the fans are installed in series, the pressure rise is doubled instead. However, the fans cannot be installed right next to each other in order to ensure more ideal operation. It should be also noted that the age of the fan might impact the emitted noise as the bearings gradually wear out.

2.7.2 Chokes

Choke is an inductive power component with a metal core which are included in frequency converters to reduce the harmonics from the output current and voltage i.e. to improve the quality of the current and voltage. If the harmonic distortions grow large enough, they can damage equipment and cause losses in supply network. [1]

The noise caused by the chokes is contributed by the phenomenon where they dissipate energy into surroundings by changing their shape and volume as a function of magnetization. This phenomenon is called magnetostriction which causes radiation of sound into the surrounding media [21, 22]. Noise caused by magnetostriction is difficult to suppress directly in drives as it requires extensive mechanical design. However, these peaks can be mitigated by using the diode mode of the liquid-cooled drive.

2.7.3 Cabinet

Sound can propagate through the cabinet material in transverse waveforms in addition to the longitudinal waves as discussed in section 2.4. As the sound waves are transmitted into the structure of the cabinet, it propagates as bending waves and finally radiates into the surroundings. However, the cabinet of a generic device is built from very thick, sturdy plates that mitigate resonance. As a result, the cabinet itself does not significantly contribute to the noise level as the noise output is dominated by the inlet fans.

The cabinet design, however, does affect the noise level radiated by the fans. The grill in front of the inlet fans serves also a noise-suppressing function. The IPXX rating of the cabinet describes the protection level against

particles and liquid. With the default IP22 filtering, the air inlet and outlet gratings are covered with metallic gratings. With the IP42 filtering, there is a mesh or a brace in between the outer and inner gratings. The IP54 filtering increases the distance of the inner and outer gratings as air filter mats are installed between the gratings. [3] Generally, the higher the IPXX filtering, the lower the sound radiated by the grill. The IPXX ratings are defined in the standard IEC 60529.

The cooling type of the drive also affects the cabinet structure, which subsequently affects the noise. In liquid-cooled drives, there are no outlet holes for the air-flow, meaning that the modules are fully enclosed. The air is circulated within the cabinet and consequently cooled with a coolant that is run through the cabinets in tubes. The lack of outlet holes decreases the overall sound power level and shifts the weight of the radiated sound spectrum.

Chapter 3

Methods

3.1 Measurement equipment and test environment

3.1.1 Measurement microphones

Microphone is a device that presents the changes in instantaneous sound pressure as a function of voltage amplitude. In microphones designed for measurements, or whenever accuracy is of utmost concern, the frequency response should be as flat as possible and it should cover the frequency range of human hearing [8]. Obviously, the microphone shall modify the original signal as little as possible.

The most common types of microphones include condenser microphones and dynamic microphones. Condenser microphones are widely used in measurements, and they are very accurate due to their wide frequency range, flat frequency response and fast transient response [12]. Condenser microphones are also very stable and have a flat frequency response. They do not significantly distort the signal as they produce little noise [12, 14].

The condenser microphone is composed of two diaphragms, one static and one thin movable diaphragm. The movable diaphragm has low mass and it moves easily by the local changes in air pressure in presence of sound waves. As both diaphragms have electrical charges, this movement causes a change in the voltage of the system as the capacitance of the system changes. The output signal of the condenser microphone is then amplified with an impedance converter. [8, 10] One disadvantage of the condenser microphone is that it is quite sensitive to moisture and thus limitations rise on the measurement environment [12].

The operation of a dynamic microphone is based on the phenomenon

of electromagnetism. A moving coil microphone is a type of the dynamic microphone where a coil is attached to a thin membrane. The coil moves inside a magnet as sound waves hit the membrane, and the changing magnetic field produces a current in the coil through induction. They do not excel at accuracy like the condenser microphones but are portable as they do not require external voltage. [8]

The directionality of the microphone is also of concern when selecting a microphone for certain applications. Directionality describes how sensitive the microphone is to different directions. Due to this property, microphones can be categorized as follows: omnidirectional pressure microphones, dipole microphones, and cardioid microphones that are front-biased. [8] The disadvantage of an omnidirectional pressure microphone is the degraded accuracy if the wavelength of the sound wave is close to the diameter of the microphone [12].

In this thesis, we use cardioid condenser microphones for their availability, accuracy and frequency range. Additionally, as the condenser microphones distort the signal the least, it is considered to produce most reliable values for engineering grade noise measurements. Transient response is not an important aspect in this thesis, as the equipment under test is unlikely to radiate transient noise.

Microphone calibration is generally done before and after the measurement to validate the measurement. It is also required by standards [13, 23]. Sound calibrators produce a signal with known parameters, and the parameters of the microphones are adjusted to this level [12].

3.1.2 Frequency filtering and A-weighting

In both one-third octave band and octave band filters, the frequency bands are equally wide on a logarithmic scale corresponding to the center frequencies of the bands. This enables a consistent frequency resolution across the logarithmic scale. In practice, this is achieved with infinite impulse response (IIR) filters. [12] The octave band and one-third octave band filters are defined in standard IEC 225.

The frequency limits and the mid-band frequency of the octave band filter are given by [12]:

$$f_0 = \sqrt{f_1 f_2}, \quad f_2 = 2f_1 \quad (3.1)$$

where f_0 is the mid-band frequency, f_1 is the lower edge frequency and f_2 is the upper edge frequency. Bandwidth is given by [12]:

$$B = f_2 - f_1, \quad B/f_0 = f_1/f_0 = 1/\sqrt{2} \quad (3.2)$$

The frequency limits and the center frequency of the one-third octave band filter are determined in same fashion excluding the upper edge frequency [12]:

$$f_2 = (2)^{1/3} f_1 \quad (3.3)$$

Frequency weighting is used to represent the effect of human auditory system on linear sound level values. These weightings are divided into A-, B- and C-weighting filters. [10]

A-weighting corresponds to the equal loudness curves discussed in section 2.5. B- and C-weighting filters are rarely used as A-weighting corresponds sufficiently to harmful sound power levels. It can also be used to estimate the irritability of the noise source [12] A-weighting is applied to the measured noise levels by adding a correction value to the sound levels in corresponding frequency bands. These corrections are defined in standard IEC 651.

3.1.3 Sound level meter

Sound level meter consists of a pressure microphone, preamplifier, windshield, detector, and filters. The equipment is used to integrate the measured values throughout the measurement to obtain the RMS value. [12]

Time weightings S for slow, F for fast, and I for impulse are used to average the RMS value according to the type of the sound signal [10]. Generally, fast time weighting is used as it corresponds the best to the human hearing [12]. Slow time weighting is sometimes used for continuous, deterministic noise [10]. Impulse time weighting is widely regarded as a niche, and the time constant has not found its use. [12].

The operating range of a sound level meter is usually from 2 Hz to 20 kHz, but the range can be increased in state-of-the-art models. The sound level meter can be typically used to obtain A-weighting and C-weighting values. [12] The classification of sound level meters is defined in standard IEC 651.

3.1.4 Sound intensity probe

Sound intensity probe is a device that is used to scan the surface of the noise source to obtain the sound intensity radiated by the said source. The probe measures the sound pressure gradient between two pressure microphones,

from which the particle velocity can be approximated. The sound intensity can be subsequently calculated with the approximation and the average sound pressure [24].

The probe consists of a phase-matched condenser microphone pair, dual-preamplifiers to eliminate the effect of vibrations on phase characteristics of the pressure microphones, and a handle that houses remote control and connections for sound intensity analyzers. The data from the microphone pair is fed into two channels. These channels can be used to determine in which direction the acoustic wave front is traveling as the leading microphone is first to interact with the wave front. [25] Sound intensity analyzer is a portable processor that postprocesses the data from the measurements into a presentable form [26].

Several interchangeable spacers that are placed in between the microphone pair are usually included with the probe kit to ensure engineering grade accuracy. The spacers hold the microphones at a certain distance from each other. The sound waves enter the microphone through the gap between the spacer and the microphone. This design improves the acoustic separation of the microphones in order to minimize cross-talk and shadowing. [25] Spacers are used to improve acoustic resolution in different frequency ranges; longer spacers are used to match longer wavelengths and vice versa [14].

3.2 Measuring the noise levels

In this section, two different standards are examined: ISO 3744 and ISO 9614-2. Sound power levels for the noise source in both octave and one-third octave frequency bands can be determined following either of these standards. However, the methods introduced in the standards are different considering the aspects of repeatability, cost, and facilities. Although not exclusively required by either of the standards, it is assumed in this study that there is an access to a hemi-anechoic room. In addition, these standards can be followed to obtain said values with grade 2 (engineering) accuracy. Grade of accuracy is defined in standard ISO 12001.

Generally, ISO 3744 is used in contrast to ISO 9614-2 as it requires less effort and is easier to repeat. However, the methods introduced in the former standard requires a costly anechoic room or a similar measurement environment, and it might be difficult to follow the standard when measuring noise levels for large devices. In free-field conditions, sound pressure level differs very little from the sound intensity level due to lack of reflecting surfaces. Consequently, results from sound pressure measurements depend on the characteristics of the room as reflections increase the local sound pressure

[14]. For this reason, the results from sound pressure measurements are not comparable if the measurement is not conducted in free-field.

Measuring sound intensity in contrast to sound pressure obtains data from the sound source while disregarding the acoustic properties of the measurement environment. In ISO 9614-2, sound intensity can be measured by scanning the noise source with a sound intensity probe. As discussed in chapter 2, sound intensity is a vector quantity given by the sound pressure and the particle velocity. In other words, we measure a vector quantity instead of a scalar quantity when measuring sound intensity. As a result, this method can be used in presence of extraneous noise sources, given that they generate steady noise [13]. Total sound power level can be easily calculated with sound intensity and, consequently, average sound pressure level can be calculated with the total sound power level of the noise source [14].

3.2.1 ISO 3744

The methods defined in standard ISO 3744 rely on multiple pressure microphones that measure the sound power level produced by the noise source on a hypothetical measurement surface. ISO 3744 is the relaxed version of the precision class standard ISO 3755 and has fewer requirements for the test environment. [14, 23]

The noise source is installed on the reflecting plane as it would be installed in field conditions. In this thesis, the generic drive is considered to stand freely on the reflecting plane away from the walls. The operating conditions of the noise source during the measurement are stated in the noise test code. The operation of the source shall be stabilized in the selected conditions before the measurement. Typically, 9 to 10 microphones are used for the measurement, but additional microphone positions might be needed depending on the size and geometry of the equipment under test. The microphones are placed uniformly on a hypothetical measurement surface that is positioned at a certain distance from the noise source. [23]

Reference box is a hypothetical right parallelepiped box that fully encloses the noise source. As a result, the reference box can have arbitrary dimensions given that they are larger than the dimensions of the noise source. The characteristic source dimension can be derived from the dimensions of the reference box by [23]:

$$d_0 = \sqrt{(l_1/2)^2 + (l_2/2)^2 + l_3^2} \quad (3.4)$$

where l_1 , l_2 , and l_3 are the dimensions of the reference box. This applies when the origin is situated on the reflecting plane and in the middle of the

surface of the reference box [23].

Hemispherical measurement surface

The characteristic dimension determines the minimum radius of the hemispherical measurement surface as the radius must be at least twice the length of the characteristic dimension [23]. This requirement sets boundaries on the measurement environment as the microphones must be positioned sufficiently far away from the walls. The microphone array on a hemispherical surface may require potentially large facilities, but it also requires the least amount of microphone positions. The microphone positions are defined in the standard. Additional microphone positions may be needed if the noise source displays prominent directivity or if the noise output of the surface varies strongly [23].

Parallelepiped measurement surface

If setting up pressure microphones on a hemispherical measurement surface is not possible due to insufficient space, they can be positioned on a parallelepiped measurement surface instead [23]. This approach can be used in relatively small spaces, but it requires a larger number of microphones. The measurement surface is given by [23]:

$$S = 4(ab + bc + ca) \quad (3.5)$$

where

$$a = 0.5l_1 + d, \quad b = 0.5l_2 + d, \quad c = l_3 + d \quad (3.6)$$

As is the case with the hemispherical measurement surface, additional measurement positions may be needed if the noise source displays prominent directivity or if the noise output of the surface varies strongly [23].

Determining the sound pressure levels

The linear or A-weighted time-averaged sound pressure levels are obtained in frequency bands for each microphone position. The mean time-averaged sound pressure level for the whole measurement surface is calculated with the following equation [23]:

$$\overline{L'_{p(ST)}} = 10 \lg \left[\frac{1}{N_m} \sum_{i=1}^{N_m} 10^{0.1L'_{pi(ST)}} \right] \text{ dB} \quad (3.7)$$

where N_m is the number of measurement points and $L'_{pi(ST)}$ is the linear or A-weighted time-averaged sound pressure level in a frequency band at a measurement position i . The time-averaged sound pressure level $\overline{L_{p(B)}}$ is calculated similarly for the background noise. The background noise correction, defined in the standard ISO 3744, is applied if the difference between the background noise level and the noise level from the source is small enough. No environmental correction needs to be applied to the sound pressure level when the measurement is conducted in a hemi-anechoic room. [23]

Determining the sound power levels

The sound power level is given by the following equation [23]:

$$L_W = \overline{L_p} + 10 \lg S \text{ dB} \quad (3.8)$$

where L_p is the time-averaged sound pressure level with applied background and environmental corrections and S is the area of the measurement surface.

Uncertainty factor

The total standard deviation is used to estimate the uncertainty of measurement in sound power levels. It is given by following equation [23]:

$$\sigma_{tot} = \sqrt{\sigma_{R0}^2 + \sigma_{omc}^2} \quad (3.9)$$

where σ_{R0} describes the repeatability of the measurement, and σ_{omc} is the standard deviation describing the uncertainty of the operation of the noise source. Both values are in decibels. The value for the standard deviation σ_{omc} is determined by repeating the measurement. For each measurement, the noise source under test is remounted and reset to the operating condition. For the repeatability of the measurement σ_{R0} , typical upper bound values can be used. [23] These values are defined in the standard.

The measurement uncertainty factor in decibels is given by the following equation [23]:

$$U = k\sigma_{tot} \quad (3.10)$$

where k is a coverage factor.

Criteria

To declare conformity to the standard, strict criteria are stated regarding the background noise, environmental noise, and measurement environment.

These criteria are defined in the standard, and they shall be satisfied for each frequency band. It must be stated whether or not the measurement conforms to the criteria stated in the standard.

The test environment shall be sufficiently isolated from background noise, so that the measurement environment is an acoustic free-field over a reflecting plane. The amount of any additional reflecting objects or sound sources shall be minimized. If ensuring acoustic free field is not possible, environmental correction can be used to an extent defined in the standard. Environmental effects, such as rain or wind, on the measurement microphones shall be minimized. For hemi-anechoic rooms this correction is zero. [23]

Additionally, the reflecting plane upon which the noise source stands should extend 0.5 meters beyond the hypothetical measurement surface and it shall have an absorption coefficient less than 0.1 for each frequency band. Apart from the environment, the standard also states criteria for the measurement equipment, including microphones and sound calibrators. [23]

Report

The standard defines the data that shall be presented in the report. First of all, the date, time and place must be recorded. For the noise source, a description shall be given on the source, auxiliary equipment, operating and mounting conditions, measurement time intervals, and mounting location within the measurement environment. If the noise source exhibits prominent directivity, additional information on the source is given as stated in the standard. [23]

For the measurement environment, a general description shall be given on the environment. In addition, the acoustical qualification and the prevailing environmental conditions shall be recorded. For the measurement equipment, the relevant data, such as the serial number, the manufacturer, and calibration dates are presented. [23]

The acoustical data that the report must include is also strictly defined in the standard. This data include the dimensions of the reference box, the shape of the measurement surface, the measurement distance, and possible background noise corrections. Most importantly measured time-averaged sound pressure levels, the surface time-averaged sound pressure levels, the sound power levels, and the measurement uncertainty factor are presented. [23]

3.2.2 ISO 9614-2

The other standard under examination for the measurements is ISO 9614-2. This standard is chosen in contrast to the standard ISO 3744 as it does not have strict requirements regarding the facilities and it is relatively simple to conduct. The advantage of measuring the sound intensity instead of sound pressure is that it is not affected by other noise sources within the measurement environment. Thus, this standard can be used for any stationary noise source in field conditions. However, sound intensity from extraneous sources and variability of said sources shall be reduced as much as practicable. The standard requires a IEC 61043, class 1 compliant sound intensity probe. The calibration of the microphones inside the probe shall be conducted with IEC 60942:2003 compliant calibrator. [13]

As defined in the standard, the surface of the noise source is divided into multiple segments. The measurement is conducted by scanning each surface segment of the noise source with an intensity probe from a specified measurement distance through a predetermined path. The probe is held perpendicular to the measurement surface with a sufficient measurement distance. The segments are selected so that each of them represent a subpart of the noise source, while taking care that the complete measurement surface fully encloses the noise source. [13]

Scanning can be either conducted manually while taking precautions not to disrupt the measurement with reflections from the body and maintaining scanning path, or with a mechanical traversing system. The scanning pattern is defined in detail in the standard. During the scanning, normal sound intensity levels are measured in all frequency bands of interest. The scanning pattern is repeated by shifting it 90 degrees. The final sound intensity is the average of these two values. The measurement is repeated for each measurement surface in each test case. [13]

Determining the sound power levels

The sound power for each segment in each frequency band is given by the following equation [13]:

$$P_i = \langle I_{ni} \rangle S_i \quad (3.11)$$

where I_{ni} is the mean segment-average normal sound intensity for the corresponding segment with area S_i . I_{ni} is given by [13]:

$$\langle I_{ni} \rangle = [\langle I_{ni}(1) \rangle + \langle I_{ni}(2) \rangle] / 2 \quad (3.12)$$

where $I_{ni}(1)$ and $I_{ni}(2)$ are given by [13]:

$$I_{ni} = I_0 (10^{I_m/10}) \quad (3.13)$$

$$I_{ni} = -I_0 (10^{I_m/10}) \quad (3.14)$$

where I_m is the measured normal sound intensity level in decibels. Equation (3.14) is used if the measured sound intensity level is negatively signed. Finally, the sound power level for each frequency band is calculated with [13]:

$$L_W = 10 \lg \left| \sum_{i=1}^N P_i/P_0 \right| \text{ dB} \quad (3.15)$$

where N is the number of segments and P_0 is the reference sound power with value 10^{-12} W. The total sound power level can be calculated with the logarithmic sum for incoherent noise sources.

Criteria

To declare conformity to the standard, some criteria and field indicators are defined in the standard.

The field indicators are used to state whether the measurement conforms to the standard. The sound pressure and the particle velocity are in same phase in a perfectly absorptive environment. However, as we cannot achieve ideal free-field conditions, the components are slightly out of phase. [14]

The pressure-intensity index describes the difference between the sound pressure and the sound intensity. In other words, this index describes whether the measurement environment is closer to a free-field or a diffuse-field. [14, 27] The surface pressure-intensity indicator F_{pI} is given by the following equation [13]:

$$F_{pI} = [L_p] - L_W + 10 \lg (S) \text{ dB} \quad (3.16)$$

where S is the area of the measurement surface, L_W is the sound power level, and surface-averaged sound pressure level $[L_p]$ is calculated with [13]:

$$[L_p] = 10 \lg \left[(1/S) \sum_{i=1}^N S_i 10^{0.1 L_{pi}} \right] \text{ dB} \quad (3.17)$$

where N is the number of segments and L_{pi} is the sound pressure level for corresponding segment S_i . For the measurement instrument, the dynamic capability index of the probe L_d must be greater than the surface pressure indicator. If the surface pressure-intensity indicator exhibits a high value in

free-field conditions, the sound probe might not be adequate for engineering class measurements. [28]

The second field indicator, negative partial power indicator, describes the difference of the incoming sound power to the exiting sound power at the measurement surface [29]. The negative partial power indicator $F_{+/-}$ is given by the following equation [13]:

$$F_{+/-} = 10 \lg \left[\frac{\sum |P_i|}{\sum P_i} \right] \text{ dB} \quad (3.18)$$

The final criterion, partial power repeatability check, determines the repeatability of an individual scan by allowing the user to detect unreliable scans. For example, transient sound caused by extraneous noise sources may cause different results for the two individual scans. The partial power repeatability is determined with [13]:

$$|L_{Wi}(1) - L_{Wi}(2)| s \quad (3.19)$$

where s is the standard deviation defined in the standard.

Report

Data that has to be included in the report is extensively defined in the standard. First of all, date and location has to be recorded. Acoustic environment, including environmental conditions, extraneous sources and description of the facility, has to be recorded. Furthermore, a description on used instrumentation has to be recorded, including calibration dates, and serial numbers. Finally, relevant acoustical data, description on measurement procedure, and statement of grade of accuracy is included in the report. Guidelines for the report are given with more accuracy in the standard. [13]

3.2.3 Equipment and test environment

The scope of this study determines that a hemi-anechoic room is used to conduct the measurements, although this is not exclusively required by either of these standards. Extraneous reflective objects must be moved out of the test environment if possible.

If standard ISO 3744 is to be followed, at least 9 microphones are required. For larger devices, this number extends up to 20 microphones. Stands for mounting the microphones may be needed, if there is no existing solution for mounting in the facility. Additionally, microphone preamplifiers for each microphone and related software is required in order to conduct the measurement. The microphones and microphone preamplifier must meet the

requirements IEC 61672-1:2002, class 1 and IEC 61260:1995, class 1, respectively. The sound calibrator for the microphones shall meet the requirements of IEC 60942:2003, class 1. The calibration of the sound calibrator shall be conducted at intervals.

The setup for ISO 9614 is relatively simple as only a sound intensity probe and related software is required. The sound intensity probe shall meet the requirements of standard IEC 1043. The calibrator shall meet the requirements of standard IEC 942.

3.3 Noise control

There are various methods of noise control, many of which are also commercially available. In this section some of these methods, and possible viable options for implementation, are discussed.

3.3.1 Interfering monopoles

Considering two arbitrary time-stationary point sources, the total time-averaged sound intensity through interference is given by [7]:

$$\bar{\mathbf{I}} = [\overline{p_1(t)\mathbf{u}_1(t)} + \overline{p_2(t)\mathbf{u}_2(t)}] + [\overline{p_1(t)\mathbf{u}_2(t)} + \overline{p_2(t)\mathbf{u}_1(t)}] \quad (3.20)$$

If the point sources are incoherent, the interference intensity denoted in the second bracket is zero. By changing the relative strengths and phases of the monopoles, the produced intensity field can be manipulated. This is assumed that the monopole sources are coherent, which is rarely the case with actual noise sources. The intensity field in ideal conditions produced by two different point sources, of which one is anti-phase, is depicted in figure 3.1. [7]

In this thesis, it is studied whether the effect of changing the angular frequencies of the array of cabinet fans in an air-cooled drive has a positive impact on the radiated spectrum. The generic drive consists of multiple sub-units i.e. cabinets. Each of these cabinets have their own set of fans and independent air flow, and the operating speed of these fans can be controlled.

By changing the angular frequency of the fans in different cabinets, the strength and the phase of the fans in a single cabinet can be shifted. It is obvious that fans cannot be operated arbitrarily as it is important to maintain proper air flow in order to prevent overheating. Pressure drop can occur if angular frequencies of the fans in a single cabinet are changed relative to each other, which leads to reduced air flow. The implementation, advantages, and

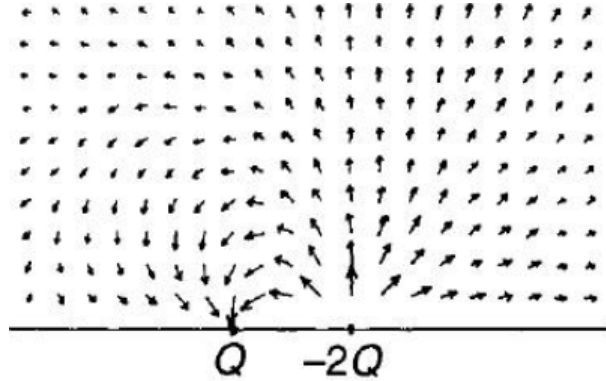


Figure 3.1: Intensity field produced by two point sources, Q indicates the volume velocity of a sound source [7].

disadvantages of monopole interference in a generic device is discussed in the next chapter.

3.3.2 Resonators

Helmholtz resonator

One seemingly favorable noise control method is the Helmholtz resonator, given that the sound propagates in a relatively narrow path. The resonator is positioned as a side branch next to this path or otherwise grazing air flow. They have been typically used for sound reproduction, but they are also useful for sound absorption especially at low frequencies as they act as reactive attenuators. [30]

The Helmholtz resonator is equivalent to a mass-spring structure that consists of a cavity of a relatively large volume acting as a spring, and a neck with a small opening leading to the cavity, acting as a mass of vibrating air. Through equivalent circuit theory, it is determined that the cavity has acoustic impedance and the neck has acoustic resistivity due to the viscous drag. These impedances differ considerably from each other due to different dimensions of the neck and the cavity. [30] In addition to these two values, radiation impedance for the opening of the neck has to be considered in practical cases [9].

There are two properties that play important role describing a Helmholtz resonator: resonance frequency and tuning frequency. Resonance frequency is a result of the impedances in the neck and the cavity that mitigate each other, creating a high acoustic pressure in the cavity. When the transmission attenuation of the resonator is at peak at some frequency, this frequency is

called tuning frequency. In an ideal resonator, these two frequencies would be the same, which is not the case in practical resonators. [30]

When the sound wave hits the tuning frequency of the Helmholtz resonator, only a part of the acoustic energy passes the resonator. Rest of the acoustic energy is transformed into reactive sound power between the sound source and the resonator. At resonance frequency, the resonator acts very differently, as the sound pressure is actually increased "after" the resonator. For this reason, it is important to make sure that the resonance frequency and the tuning frequency are sufficiently far apart from each other. [30]

The Helmholtz resonator dampens frequencies near its resonance frequency. The resonance frequency depends on the dimensions of the resonator [9]:

$$f_0 = \frac{c}{2\pi} \sqrt{\frac{S}{LV}} \quad (3.21)$$

where S and L are the cross-section and the length of the the neck, V is the volume of the cavity, and c is the speed of sound.

It is worth noticing that the resonating frequency of a Helmholtz resonator can be controlled by changing the diameter of the opening. If a neck cannot be incorporated due to limited space, the natural frequency can be calculated with the following equation that takes the effective length into account for a neckless Helmholtz resonator [9]:

$$f_0 = \frac{c}{2\pi} \sqrt{\frac{1.85d}{V}} \quad (3.22)$$

Helmholtz resonators can be used to suppress certain frequencies from the sound spectrum emitted by the noise source. It is especially useful when it is possible to identify irritating frequencies, often at the higher frequencies of the hearing range. The degree of losses decreases the attenuation at the resonating frequency and broadens the range of the attenuated frequency bands. [19]

Quarter-wave resonator

Quarter-wave resonators can be installed as side branches, similarly to Helmholtz resonators, to the path of the sound waves. True to its name, the length of the quarter-wave resonator is a quarter of the wavelength of the sound wave [19, 31]. Resonance condition for a quarter-wave resonator is determined by the following equation [31]:

$$L = \frac{(2n - 1)\lambda}{4}, \quad n = 0, 1, 2, 3 \dots \quad (3.23)$$

Where L is the length of the resonator and λ is the wavelength of the sound wave.

3.3.3 Absorbing materials

As discussed earlier, choice of material is not a trivial matter when it comes to acoustic properties. Absorbing materials can be used to attenuate the sound levels. Porous materials, for example, are effective at attenuating sound levels in mid and high frequencies. Porous materials consist of a solid structure and fluid filled pores within the structure. The interaction of the solid medium with the fluid in pores cause fluctuation of the gas flow, leading into molecular layers with different relative velocity. As discussed in section 2.2, the shear stress caused by this difference of momentum dissipates some of the sound energy. Some examples of porous materials include glass fiber and mineral wool. [7]

The main advantage of a porous material is that it does not require large volume to be effective at mid and high frequencies, which is generally not the case at low frequencies for porous materials. To attenuate low frequencies, membranes and perforated panels can be used instead. However, these membranes or panels cannot be directly applied to a surface as a layer but they require a cavity behind the material. Membranes rely on the mass of the membrane to absorb a part of the sound energy, whereas perforated panels rely on discrete, local Helmholtz resonators composed of the perforations and the cavity. [15]

As discussed in section 3.3.2, the resonating frequency of a Helmholtz resonator can be controlled with the size of the cavity and the opening i.e. the perforations. Micro perforated plates (MPP) operate similarly to perforated plates, but the perforations have smaller diameters. These thin absorbers can be used to attenuate sound levels in a wider frequency band. [32] To improve the noise-attenuating properties of the panels, different panel structures can be used instead of single thin panels. For example, coupled panels together with a cavity filled with absorbing material (such as mineral wool) is an efficient solution, but may require extensive modifications on existing structures [19].

Using materials with a repeated structure of Helmholtz resonators, the propagation of sound can be dispersed, effectively shifting the resonance frequency to low frequencies [15]. Metamaterials are such materials where the repetition of the structure is small in contrast to the wavelength. The advantage in contrast to porous materials is that metamaterials do not require thick layers to dissipate sound energy at low frequencies. Locally resonant acoustic metamaterials (LRAM) are metamaterials, with a structure consist-

ing of discrete resonators, designed to attenuate certain frequencies. Such materials can also be used to attenuate propagation into certain direction. [33] Metamaterials generally absorb sound energy in narrowband but broadband meta-materials are currently developed [15].

In thin plates, Bragg gratings can be used to attenuate sound waves with wavelengths of certain dimension or direction. Discontinuities, such as grids or ribbed plates have been observed to be useful in noise control through interference. They can contribute to a stopband in the sound spectrum at wavelengths of same size of the repeated structure. [33] There are several types of commercially available metamaterials, and discussing them in detail will be out of scope of this thesis.

Although not the main concern in this thesis, structure-borne sound does contribute to the total sound power level of a frequency converter. Structure-borne sound is relatively difficult to attenuate in high-power frequency converters without affecting the mechanical design. However, elastic epoxy is one favorable option developed in collaboration by Noisetek and VTT (Technical Research Centre of Finland). The material aims to attenuate the vibration of the structure, and consequently the radiated structure-borne sound [34].

Elastic epoxy is a viscoelastic material that is applied on a small area of the surface of the radiating structure. When applied, the material resists the stresses that cause bending waves within the materials. In contrast to traditional viscoelastic materials, elastic epoxy attenuates bending waves optimally in a wider temperature range as it can be tuned to attenuate certain frequencies at certain temperatures. [34]

Chapter 4

Implementation

4.1 Overview of the measurement procedure

This section describes the measurement procedure that was used to determine the sound power level in engineering grade, both in octave bands and in one-third octave bands. We decided to follow the scanning method described in the standard ISO 9614-2:1996. Two main reasons were in favor of this decision. Firstly, the relaxed requirements on the extraneous noise sources gives more options on the environment. If required, scanning method can be used in other facilities than those designed strictly for noise tests. This is a great advantage as moving and installing drives requires a fair amount of work force and hours. Secondly, the scanning method does not have strict requirements on the dimensions of the testing facility. We found this reason to support the decision to measure sound intensity instead of sound pressure as the available testing facilities had fairly small dimensions relative to the dimensions of an average drive.

As described in section 3.2.2, the standard ISO 9614-2:1996 relies on the measurement of two quantities. Two pressure microphones are used to measure the sound pressure radiated by the surface of the noise source. In addition, the particle velocity can be calculated from the sound pressure gradient captured by the two microphones. As the advantage over just measuring the sound pressure, we can discern extraneous sound pressure from the sound pressure radiated by the noise source of interest.

Before beginning with the measurement, it should be examined what the end result of the measurement should be. This depends on the noise spectrum that the noise source radiates. For example, it is not advantageous to measure low frequencies if the source mainly radiates in high frequencies. Furthermore, if the noise source radiates broadband noise with discrete tones,

it might be wise to measure sound intensity over a wider range of frequencies in one-third octave bands for more precise results. However, if only the total sound power level is of interest rather than the components of the spectrum, the measurement can be measured sufficiently in octave bands. In this thesis, we measure the sound intensity in both octave bands and one-third octave bands at center frequencies ranging from 63 Hz to 8000 Hz.

Selecting the measurement surface segments is not a trivial task either. Smaller surface segments make it easier to obtain accurate values, especially if the source does not radiate noise uniformly. However, more segments result in more scans, which can increase the time worked on the measurement, especially if multiple spacers are needed for the selected frequency range of interest. Changing spacers affects the frequency range of the measurement.

The measurement itself is relatively simple. Two separate scans with orthogonal scan patterns are conducted on each measurement segment. Typically, for each scan the data is processed into a single value of sound intensity. Additionally, a single value for sound pressure is obtained to determine the surface pressure intensity index. The sound intensity data is finally processed into sound power level in octave bands or one-third octave bands. Scanning speed, scanning patterns, and processing the measurement data are discussed in greater extent in section 3.2.2.

If it is not clear what level of resolution is required in the resulting sound power level spectrum, it is safer to obtain the values in one-third octave bands for greater accuracy. The resulting sound power levels can be effortlessly calculated from one-third octave band to octave band with the following equation:

$$L = 10 \lg (10^{L_1/10} + 10^{L_2/10} + 10^{L_3/10}) \quad (4.1)$$

Where L_1 and L_3 are the sound power levels in one-third octave bands liminal to the shared center frequency in both octave bands and one-third octave bands with the sound power level L_2 .

4.2 Measurement environment

We conduct the measurements in a hemi-anechoic room with dimensions of approximately 8 m x 8 m x 8m. The dimensions are long enough to follow the standard ISO 9614-2:1996. However, a hemispherical measurement surface stated in the standard ISO 3744 would not fit into such space. A parallelepiped measurement surface requires greater number of microphones for such a source, and the testing facility was not fitted to easily incorporate additional microphones. In addition, a sound intensity probe kit, described in the next section, was readily available at the time of the measurement.

As discussed earlier, we deemed the sound intensity measurement as a more sensible approach.

The number of reflecting surfaces are minimized during the scan. Precautions are taken so that the user of the probe would not act as a reflecting surface.

4.3 Measurement equipment

Following the standard ISO 9614-2:1996, compliant equipment is selected to conform to the standard. A sound intensity probe kit includes a wind-screen, the probe, a sound intensity microphone pair, a dual pre-amplifier, and usually multiple spacers. The sound intensity is calculated by the sound intensity processor from instantaneous pressure and particle velocity that is further obtained from the pressure gradient. For this task we use a computer software coupled with an input module for the probe. The sound intensity probe kit complies with the standard IEC 1043, class 1. [25]

4.3.1 Probe

In this thesis, we use a sound intensity probe manufactured by LMS (GRAS), type 50AI-LP. The sound intensity probe is designed to hold the pressure microphones and the dual preamplifier. According to the manufacturer, the measurements with the probe kit are valid in center frequencies ranging from 60 Hz to 10 000 Hz, which conforms to the requirements of the measurements. The sound probe is held with an extension handle with a 12-pin LEMO connector plug which is further used to connect the probe to the sound analyzer, or in our case to the input module. The control buttons in the handle enables the user to begin and end each scan remotely. [25]

4.3.2 Sound intensity microphone pair

The probe Type 50AI-LP includes a phase-matched 1/2" microphone pair of type 40GK, also manufactured by GRAS. The probe kit is also fitted with 1/4" preamplifiers. [25]

The probe kit includes three spacers with lengths 12 mm, 25 mm, and 50 mm. In this thesis, we follow the guideline provided by the manufacturer of the probe. Based on figure 4.1, we use a 50 mm spacer to cover center frequencies ranging from 63 Hz to 1250 Hz and a 12 mm spacer for center frequencies ranging from 1600 Hz to 8000 Hz.

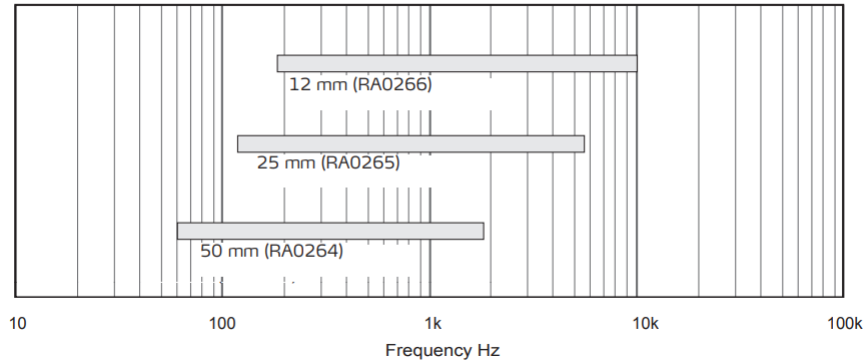


Figure 4.1: Spacer sizes and related frequency ranges [25].

4.3.3 Sound intensity analyzer

We did not have access to a dedicated sound intensity analyzer during the measurements. Instead, a 12-pin LEMO with BNC jacks were used to connect the two microphone channels to a 6-channel input module. The input module, Type 3050-B-060 manufactured by B&K, was used as an interface for the measurement and analysis software to obtain the measurement data. The input module allows for real-time data acquisition in multiple channels [35]. PULSE software developed by B&K is used for measurement data analysis.

The setup introduced a few disadvantages, including the requirement of additional person to operate the software, and the decreased mobility as the computer was required to conduct the measurements. Furthermore, the lack of sound analyzer dedicated for sound intensity increased the lead time of the measurement process as the data had to be further post-processed after exporting the data from the measurement software.

For other than sound intensity measurements, we use a sound level meter Type 2250 manufactured by B&K to obtain acoustic data.

4.4 Test cases

The test cases are divided into two main cases: the air-cooled drive and the liquid-cooled drive. The reasoning was that both cooling types affect the radiated noise spectrum differently. The tests were conducted for cases where the EUT was driving a load, i.e. a motor. For this purpose, we drive a 1 MW motor with test cases involving lower output currents and a 2 MW motor with test cases involving higher output currents. Initially, several more test cases were scheduled. However, the lead time was dramatically increased

throughout the process, reducing the total number of test cases. The test cases that were left out from this thesis are further discussed in the next chapter.

Most of the results for the test cases have been obtained with the scanning method for sound intensity. However, some of the tests have been conducted with a hand-held sound analyzer. The used measurement equipment is mentioned for each test.

4.4.1 Air-cooled drive

The first drive we test is a typical air-cooled drive. The EUT consists of the drive unit, supply unit, income and output units for cabling, and an auxiliary control cabinet. However, a few changes were made to the default design that have an effect on the obtained results. First of all, the drive is installed on a movable platform. This leads to cases where abrupt changes in the PI index can be observed when measuring near to the lower edge of the drive due to reflections from the platform. In addition, the drive was equipped with IP54 filters to protect it from harsh environmental conditions. The IP54 filters have a noticeable effect on the noise level as discussed later. Preliminary examination indicated that the main noise source was the inlet fan located in the drive unit.

The measurement surface segments used during the measurement are depicted in figure 4.2. The measurement surface was divided to incorporate similarly radiating areas into one surface segment. If any of the subareas of a segment radiate strongly in relation to another subarea, it could manifest as a higher value for PI index. For example, the inlet grills are measured separately as they output noise at a higher level than the surrounding areas. In this drive, the most interesting segments include segments 1-6. Supply inlet grill and inverter inlet grill are located in segments 2 and 3, respectively.

First of all, we determine the sound power level in octave bands and the total A-weighted sound power level at typical operating conditions, meaning that the current output is similar to field conditions. However, parameters are altered to operate the fans at maximum speed to simulate the worst case scenario with the respective output current. Secondly, we determine the same values in the worst case scenario. The worst case scenario means that the fans are driven at their full speed at operating point similar to field conditions. In addition, we explore how changing the fan speed and the grill design for these fans affect the noise spectrum radiated by segment 3.

When examining how fan speed affects the noise level, we measure the noise level at multiple operating points. As it is not likely that the fan speed can be decreased drastically to prevent overheating, we aim to reduce the

speed at small intervals to see if we can add the noise from the inverter unit destructively with the noise from the supply unit.

The noise-suppressing grill is simple in design. Similar to other noise-suppressing air ventilation grills, obstacles covered with noise-absorbing material are installed inside of the aluminum frame. This leads to a structure similar to lamella silencers that are used directly next to fans in other applications. As a result, the air-flow through the inlet grill is rerouted to increase the travel distance of the flow. Meanwhile, the 10 mm layer of absorbing material that is installed throughout the grill suppresses the vibration of the air. As the absorbing material, a commercially available Basotect was used for its non-toxicity and fire-resistance [36]. These two properties are of utmost concern as the currents that are in play with high power drives can cause significant heat inside the cabinet.

The decision to measure in octave bands resulted from the fact that there were no tonal components to be heard in the radiated noise. In addition, it was easier to monitor the criteria that confirm the grade of accuracy of the measurement as stated in the standard ISO 9614-2:1996 as conformation to these criteria had to be confirmed in post-processing in our measurement setup.

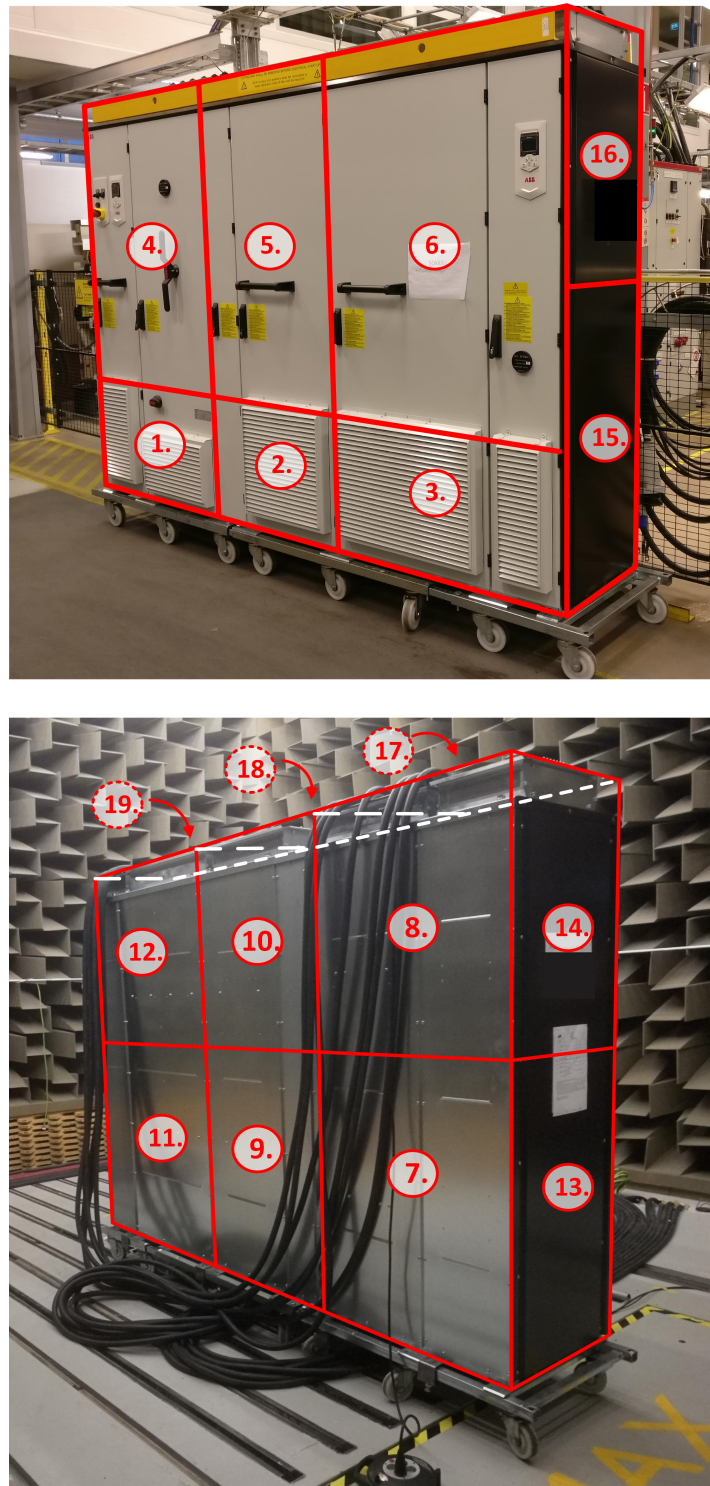


Figure 4.2: Measurement surface segments for the air-cooled drive; segments 1-3 contain the inlet grills of the drive.

4.4.2 Liquid-cooled drive

The second EUT we measured during the noise tests is a typical liquid-cooled drive. The drive consists of a drive unit, a supply unit, a line filter unit, and a cooling unit. As was the case with the air-cooled drive, the drive is installed on a movable platform which may cause similar problems when determining the PI index. Preliminary examination indicated that the noise spectrum contains tonal components as there was a distinct tone to be heard.

The measurement surface segments used during the measurement are shown in figure 4.3. As was the case with the air-cooled drive, the measurement surface was divided to incorporate similarly radiating areas into one surface segment. However, it is assumed that the drive radiates noise quite uniformly on side, hence the measurement surface is divided into larger segments.

We noticed that better resolution is needed to determine the tonal component of the noise spectrum, and for this reason we decided to measure the data at one-third octave bands. First, we determine the sound power level in one-third octave bands and the total A-weighted sound power level at different operating points. These operating points determine the output current of the drive. The operating points range from low output current to full load (0 % to 100 % of rated current). Additionally, the drive is tested at the full load operating point with the diode mode on to determine how it affects the tonal components introduced by the coil through magnetostriction. The diode mode is essentially a control program for the diode supply unit that automatically switches the main contactor on and off [37].

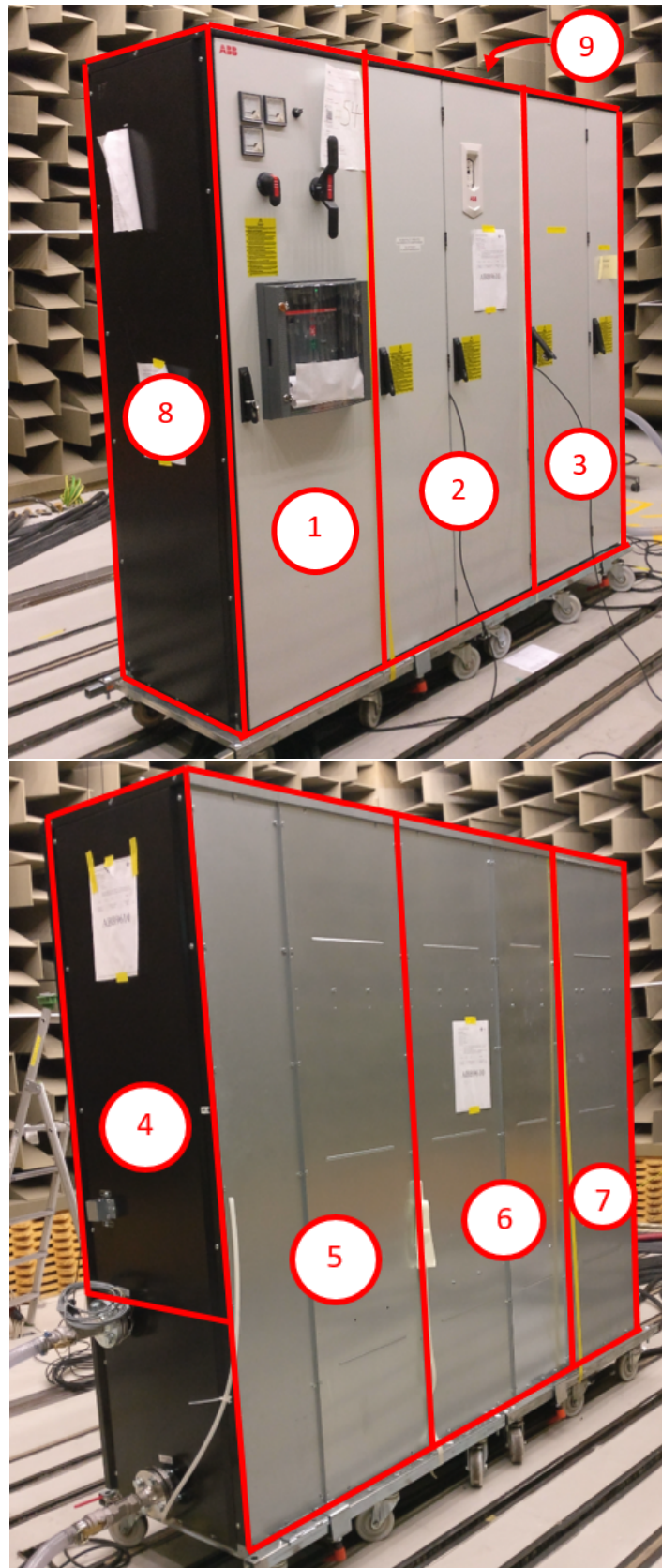


Figure 4.3: Measurement surface segments for the liquid-cooled drive.

Chapter 5

Measurement results

In this chapter, the obtained results are presented and discussed for each test cases described in chapter 4. An overview of the test cases is shown in the table 5.1. This table is used to reference to the test cases in this chapter.

Test	Drive type	Description	Output (A)	Voltage (V)
1	AC	Typical operation	1010	690
2	AC	Worst case	1010	690
3.1	AC	Fans, 100 % & 100 %	1010	690
3.2	AC	Fans, 100 % & 95 %	1010	690
3.3	AC	Fans, 100 % & 90 %	1010	690
3.4	AC	Fans, 100 % & 85 %	1010	690
3.5	AC	Fans, 85 % & 85 %	1010	690
4.1	AC	Grill, IP42	1540	690
4.2	AC	Grill, IP54	1540	690
4.3	AC	Grill, noise suppression	1540	690
5.1	LC	Output current, 0 %	460	690
5.2	LC	Output current, 30 %	540	690
5.3	LC	Output current, 45 %	723	690
5.4	LC	Output current, 75 %	1245	690
5.5	LC	Output current, 100 %	1660	690
6	LC	100 %, diode supply	1660	690

Table 5.1: Overview of the test cases; AC = air-cooled, LC = liquid-cooled

Additional test cases were initially scheduled to the time slot reserved for the measurements. These test cases include several operating points for the air-cooled drive, equipping the inlet fan sleeves inside the cabinet with noise-suppressing grills, and using different noise-absorbing materials to cover the

inside of the cabinet. We had sourced two materials for tests: Basotect and Ecophon Advantage A.

Basotect is lightweight, reworkable, and foam-like porous material manufactured from melamine resin. It is designed particularly to absorb high frequencies. For low frequencies, thicker layers are required, which might not be achievable in a generic drive. The material is non-flammable and can be used in applications with temperature up to 240 Celsius. [36]

The latter material is a non-reworkable acoustic tile with the thickness of 15 mm, designed for building acoustics designed to absorb acoustic frequencies ranging from 250 Hz to 4000 Hz. The material is light and fire-resistant as it is manufactured from glass wool and it withstands heat up to 300 Celsius. [38] However, as we noticed that covering the inside of the door with relatively thin Basotect caused serious problems with the air flow, we decided not to test Ecophon Advantage A either. If more time would have been available for additional test cases, covering the cabinet doors with a thinner layer of reworkable Basotect would have been tested.

Due to several setbacks with measurement equipment (missing or broken) and installation of EUTs, the time reserved solely for the measurements was drastically decreased from weeks to days. Something that also added to the lead time of the project was the fact that measuring frequency converters with load in a hemi-anechoic room is quite a novel idea that had not been done previously for several years.

5.1 Typical operating conditions (Test 1)

The test was conducted for the air-cooled drive with the sound intensity probe following the sound intensity scanning method. The spectral content of the radiated noise is depicted in figure 5.1, and the sound power level per segment is depicted in figure 5.2

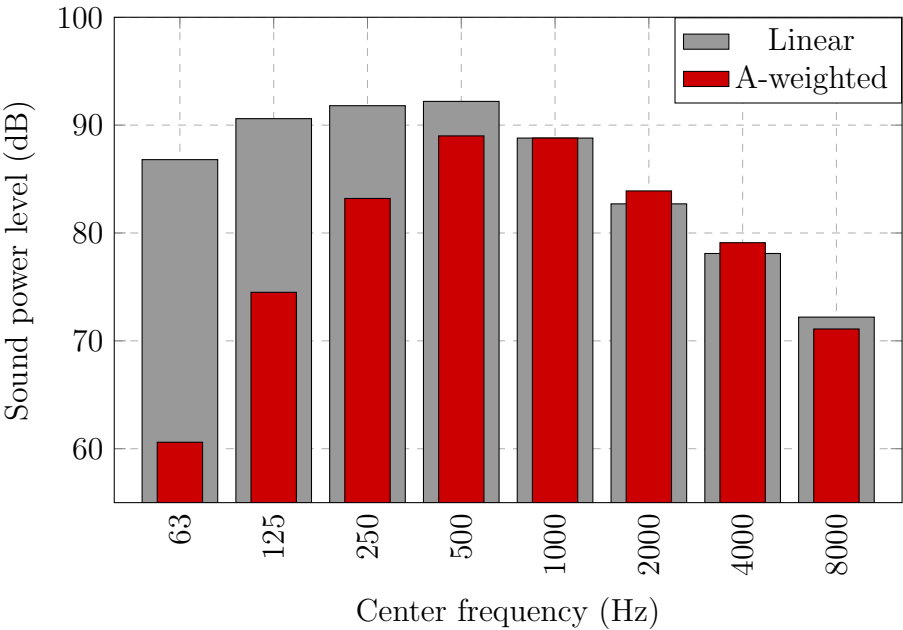


Figure 5.1: Test 1. Sound power level spectrum in octave bands.

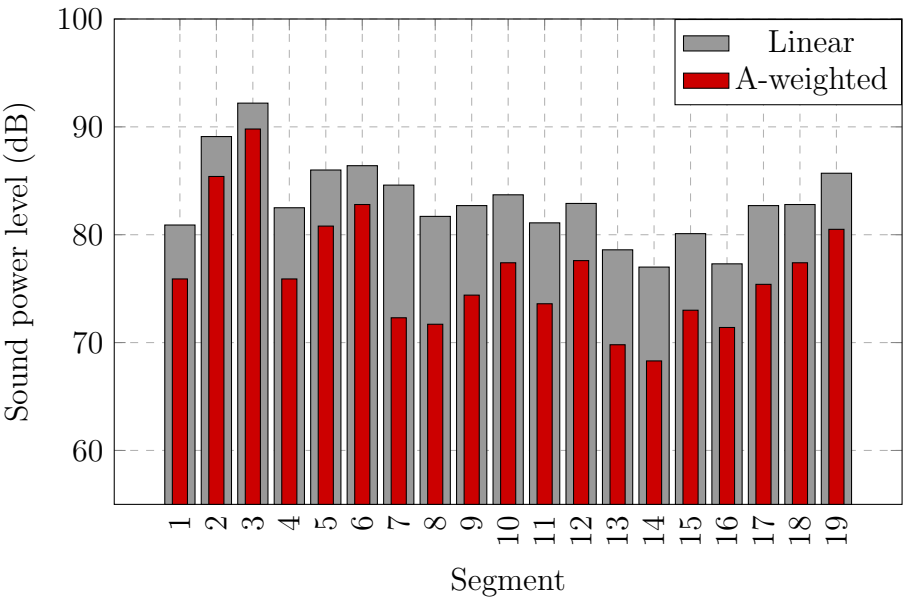


Figure 5.2: Test 1. Sound power level spectrum per segment.

As it can be seen from figure 5.1, the A-weighting emphasizes the sound power level at 2 kHz and 4 kHz. In frequency bands ranging from 500 Hz to 8

kHz, the sound power levels are close to the linear values. In contrast, the A-weighted sound power levels at lower frequencies are dramatically reduced. As a result, lower frequencies do not significantly contribute to the total A-weighted sound power level.

In figure 5.2, it can be seen that A-weighted values in the front, especially supply and drive side inlet grills in segments 2 and 3, are higher compared to the rest of the drive. This is the expected outcome as the inlet fans contribute the most to the noise output in mid and high frequencies. The segments in the back (7-12) have relatively lower levels as the gravity of their relevant spectra is shifted towards low frequencies.

The measurement conforms fully to the standard, passing all the requirements for field indicators and the partial power repeatability check.

5.2 Worst case operating conditions (Test 2)

The test was conducted for the air-cooled drive with the sound intensity probe following the sound intensity scanning method. The spectral content of the radiated noise is depicted in figure 5.3, and the sound power level per segment is depicted in figure 5.4.

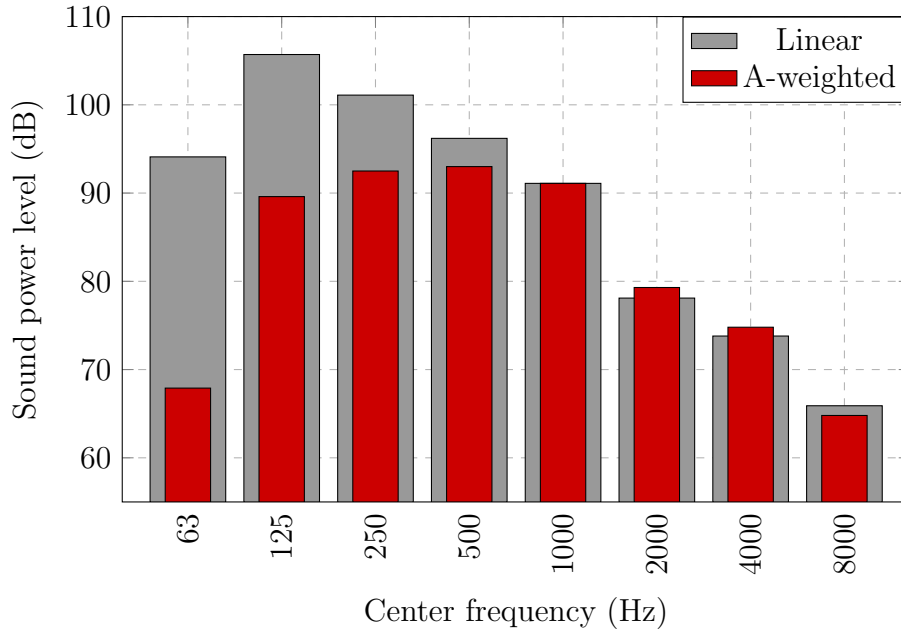


Figure 5.3: Test 2. Sound power level spectrum in octave bands.

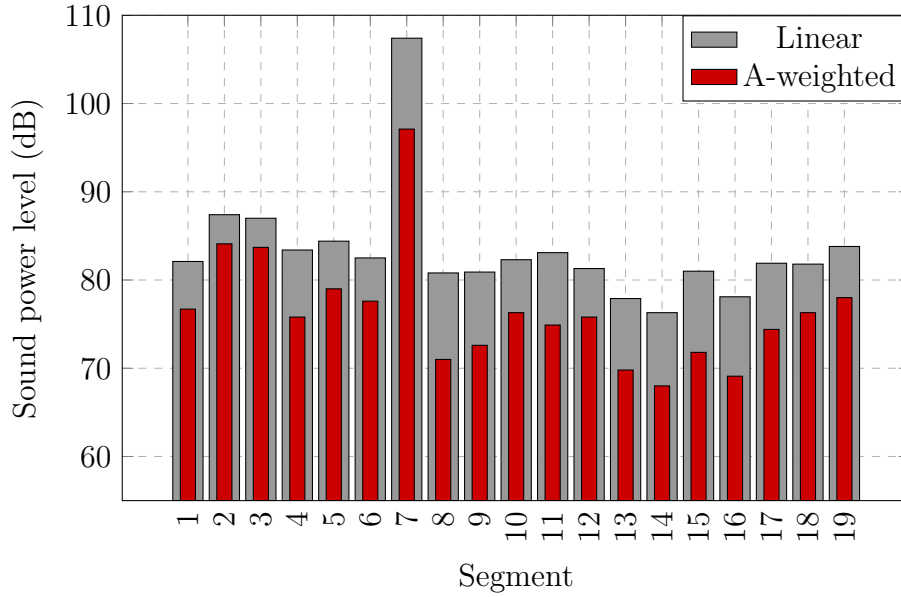


Figure 5.4: Test 2. Sound power level spectrum per segment.

When examining the spectra (figures 5.3, 5.4), abrupt peaks can be noticed to stand out. These are not tonal characteristics, but rather errors in the measurement procedure. This was also proven by examining the field indicators calculated according to the standard ISO 9614-2:1996. These errors render the obtained value for total sound power level useless.

There are a few factors that could have caused this error. The measurements included to this test case were first in chronological order. The sound intensity probe users did not have prior experience to the scanning method, resulting in an unacceptable variation in scanning speed and probe orientation. These factors are easy to see in field indicators that are calculated when post-processing the measurement data. Many of the individual scans failed to meet the criteria given in the standard. Secondly, this is the only test where we did not have the access to the microphone pair calibrators. However, this did not likely affect the results significantly as the gain corrections differed very little prior to and after the calibration.

Finally, the most likely factor to cause the error in addition to the inexperience of the users was the placement of the input and output cables. As shown in last chapter, the input and output cables were installed by routing the cables from the back of the drive. Upon examination, we noticed that the current in the cables cause significant error in the probe when the scanning was conducted close (about 15-20 cm) to the cable. We also noticed

significant errors if the cable between the probe and the input module was laying on these cables. Additionally, the cables are very thick and they can likely cause reflections that can affect the results. We noticed that this can be prevented by increasing the scanning distance slightly. In further measurements, we took precautions for these errors. For the rest of the test cases, we also decided to only measure the area that was not covered by the cables and approximated the rest of the segment, leading to more reliable results.

5.3 Varied inlet fan speed (Test 3)

The tests were conducted for the air-cooled drive using the sound level meter as we did not have the sound intensity at our disposal at the time. The tests were not conducted following a standard; the results should not be treated as absolute values, but they can be analyzed comparatively. Instead of using sound power levels, the results are presented as A-weighted fast time weighted equivalent continuous sound levels as this was supported natively by the sound level meter. The equivalent continuous sound level describes the average sound energy during the measurement.

The results were obtained for multiple cases while decreasing the speed reference for the fans in the inverter unit and the supply unit by small intervals. Changing the speed reference is the only way this could be achieved; no absolute values could be determined to operate the fans. As we noticed that the speed reference does not operate linearly, the decrease of 5% was deemed as a small interval. The sound level meter was mounted on a stand at height of 1 meter, and the mount was placed at a distance of 1 meter from the drive. The sound level meter faced the boundary of the supply unit and the inverter unit.

The results with total A-weighted equivalent continuous sound pressure levels are presented in figure 5.5. It can be seen that decreasing the fan speed by altering the fan speed reference slightly does not have a drastic effect on the noise level. The inlet fan grill on the inverter unit remains as the main source to the noise output and no negative interference is achieved.

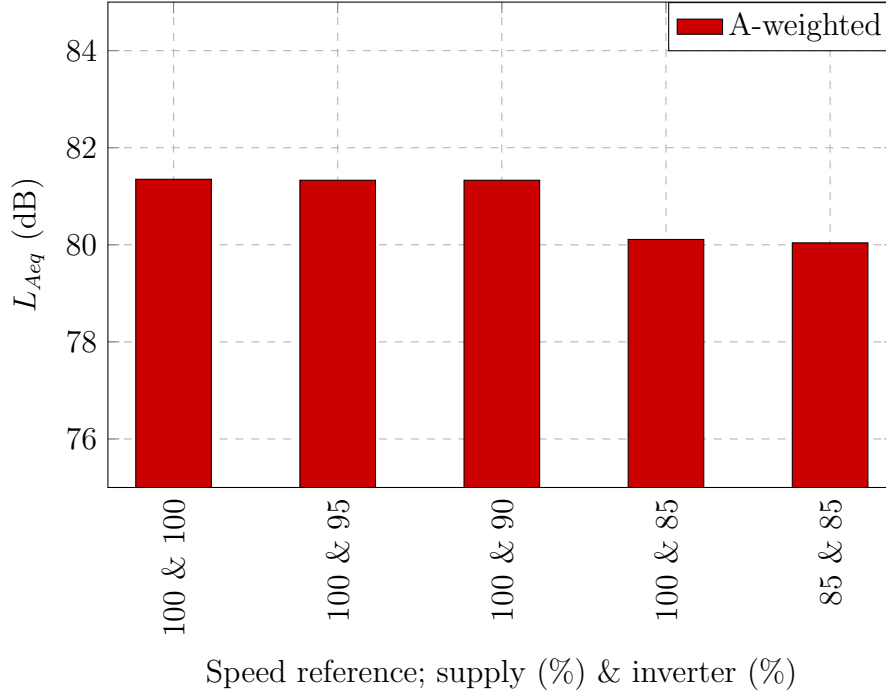


Figure 5.5: Test 3. Total A-weighted equivalent continuous sound pressure level.

5.4 Varied inlet grill structure (Test 4)

The test was conducted for the air-cooled drive by scanning the inlet grill of the inverter unit with three different cases. These cases include the default IP54 filtering and without IP54 filtering, and switching the default grill to a prototype that is specifically designed to suppress noise. This prototype is not fitted with IP54 filtering. An overview of the structure of the noise-suppressing grill is given in previous chapter. As discussed earlier, we used Basotect as the absorbing material.

In figure 5.6, it can be noticed that the noise-suppressing grill design (referred as silencer) does dramatically reduce noise in higher frequencies. As expected, fluid vibrations are difficult to attenuate through absorption, and the sound power level in lower frequencies are not remarkably lower. In range 63 Hz - 1000 Hz, the sound power level with noise-suppressing design is around the same level as the default design with IP54 filter. However, the low frequencies are of little importance after applying the A-weighting corrections.

However, as the new design is not equipped with IP54 filtering it is not

suitable for use in harsher environments. Introducing IP54 filtering to the noise-suppressing design would decrease sound power levels even further. On the other hand, it was not monitored whether the temperature inside the cabin is stable when using this design. For further applications, it should be studied how much of pressure drop is introduced with such a design to ensure proper air-flow through the cabinet.

Looking back at the results of the test case with typical operating conditions, it would be interesting to see the effect on the total A-weighted sound power level. Judging from figure 5.7, the most prominent frequency bands with center frequencies 500 Hz and 1000 Hz would not be affected significantly. However, the frequency band with center frequency 2000 Hz could see a noticeable drop in sound power level. Such a test case was on our scope, but had to quickly be dropped in order to get the next EUT to the site.

The obtained results passed the requirements the surface pressure-intensity index and power repeatability check, but does not fully conform to the standard as only a part of the measurement surface is scanned. Due to limited time, these measurements could not have been conducted for the whole noise source, so the results can not be used as absolute values, but rather to make comparisons between each grill structure.

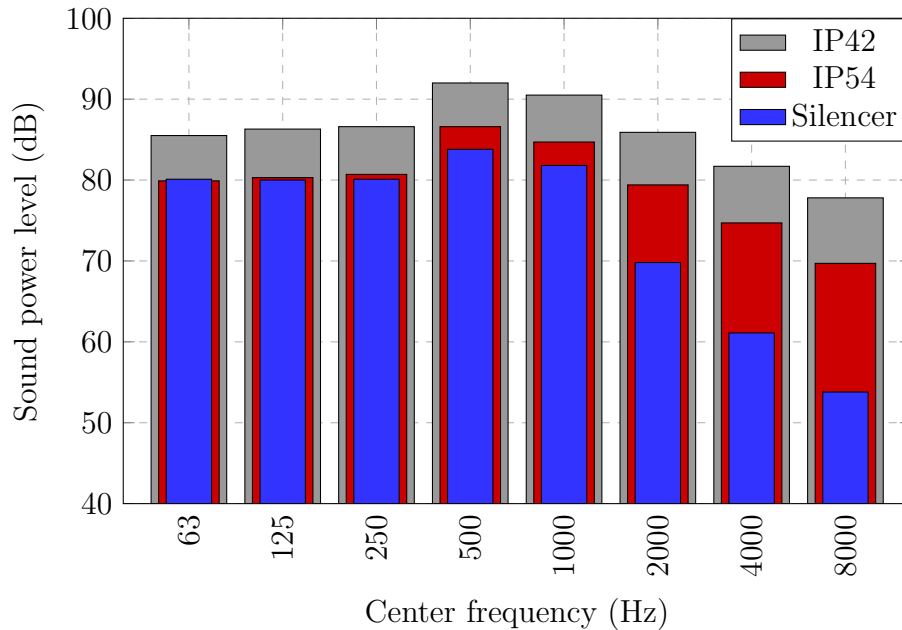


Figure 5.6: Test 4. Linear sound power level spectrum in octave bands.

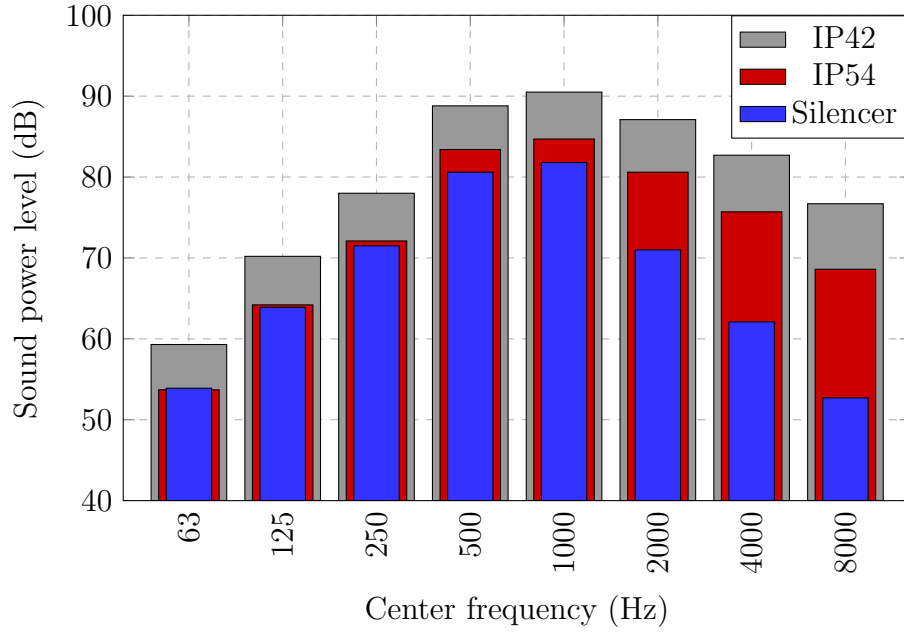


Figure 5.7: Test 4. A-weighted sound power level spectrum in octave bands.

5.5 Varied output current (Test 5)

The tests were conducted for the liquid-cooled drive with the sound intensity probe following the sound intensity scanning method. The A-weighted spectral content of the radiated noise for each test case are presented below. As it can be seen from the spectra, there are several apparent peaks located at center frequencies 315 Hz, 3150 Hz, 4000 Hz, and 8000 Hz. The peaks at 3150 Hz and 4150 Hz are most likely caused by the magnetostriction of the coil, and they persist distinctive even if the current output of the drive is increased. The largest (and the most significant) gain in sound power level as a function of output current level is at these peaks.

These peaks in the higher end of the spectrum could be attenuated with a similar kind of absorbing material that was used with the noise-suppressing grill. In fact, this was tested by covering the inner surface of the door with Basotect. The thickness of the layer was 10 mm. However, the test quickly ended as the frequency converter trips due to pressure drop caused by the material and no measurement data was obtained. In future applications, thin layers of absorbing material should be used in order to maintain proper air-flow. As a result, the sound power levels are not attenuated as dramatically than with thick layers. This test case was on our scope, but further attempts

had to be dropped as the time slot allocated to the thesis was exhausted.

The tests conform to the standard in most parts. One scan had the scanning time less than 20 seconds, but the measurement result was considered reasonable when compared with other data. A few of the scans failed the field indicator tests at lowest frequencies. However, the A-weighting mitigates the possible error as mid and high frequencies are emphasized. For this reason, the resulting total sound power level is still deemed reliable.

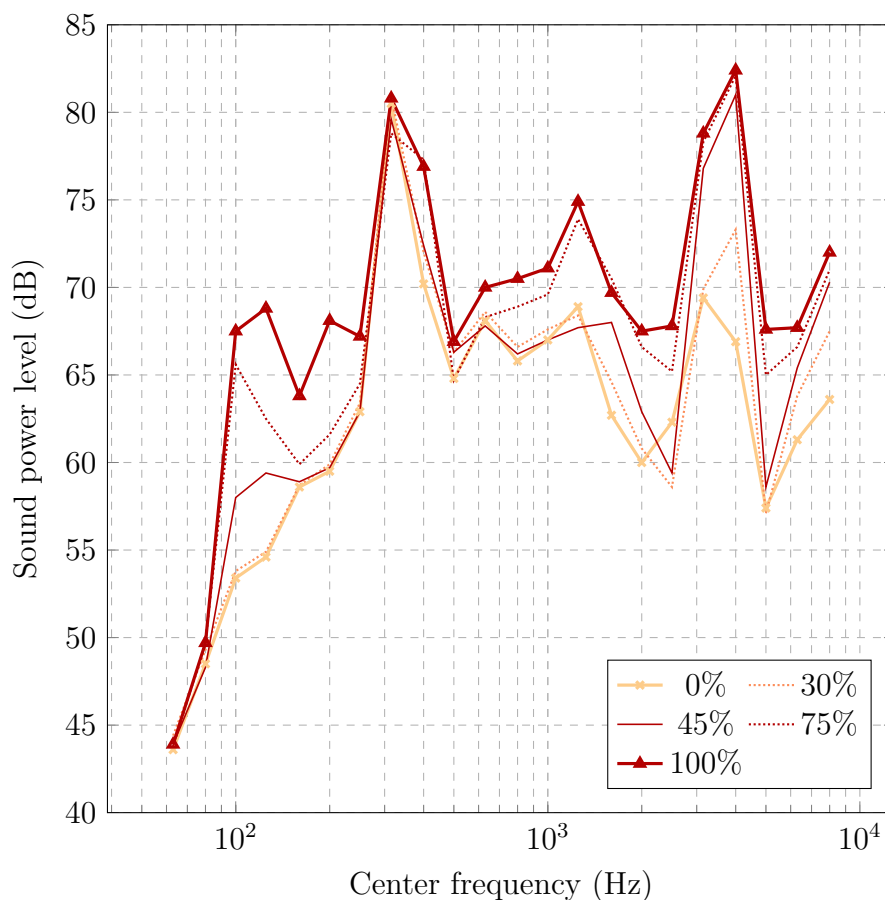


Figure 5.8: Test 5. A-weighted sound power level spectra in one-third octave bands.

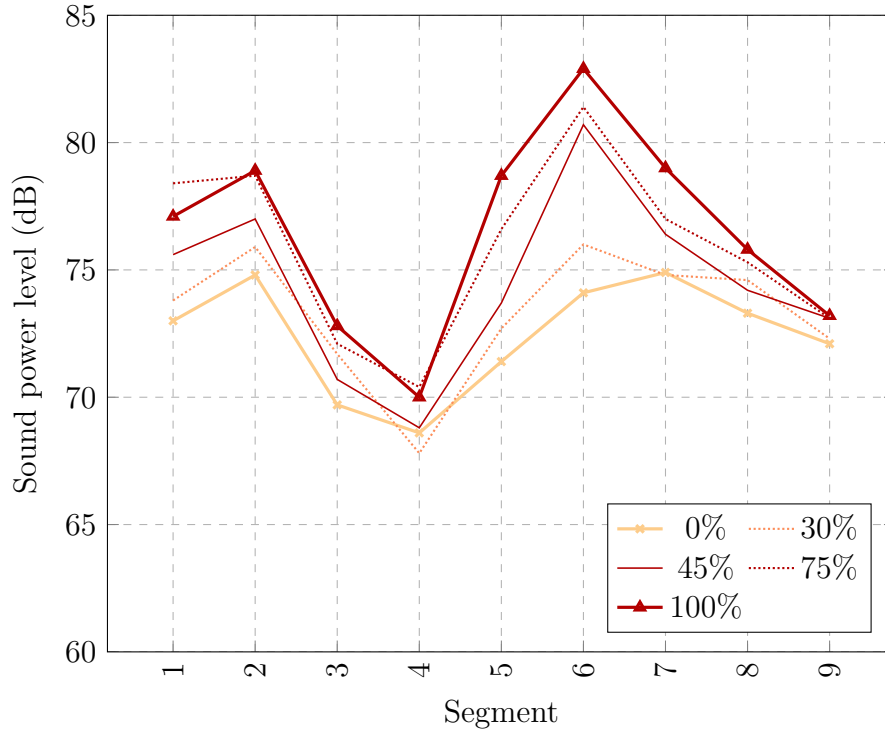


Figure 5.9: Test 5. A-weighted sound power levels per segment.

5.6 Nominal output current with diode mode (Test 6)

The test was conducted for the liquid-cooled drive with the sound intensity probe following the sound intensity scanning method. The spectral content of the radiated noise is depicted in 5.10. The obtained results conform fully to the standard, passing all the requirements for field indicators and the partial power repeatability check. The test case 5.5 has been added as reference data to both figures.

It can be noticed that the spectrum is dramatically different than with the the test case 5.5, even though same operating point is used. Most notably, the peaks at center frequencies 3150 Hz, 4000 Hz, and 8000 Hz are virtually non-existent anymore. This difference can also be easily heard by ear. In fact, the higher end of the spectrum is attenuated significantly, but some sound power levels in the lower end of the spectrum are slightly increased. As a result, the total A-weighted sound power level is not significantly lower. However, the lack of tonal characteristics makes the spectrum smoother and

the perceived sound much less irritating.

Interestingly enough, the linear sound power level radiated by each segment are higher than their counterparts without diode supply. This can be seen in figure 5.11. In contrast, the A-weighted values are slightly lower than with the test case 5.5. This is due to different spectral content as A-weighting emphasizes the mid and high-frequencies of the spectrum. As a result, the total A-weighted sound power levels differ very little from the reference.

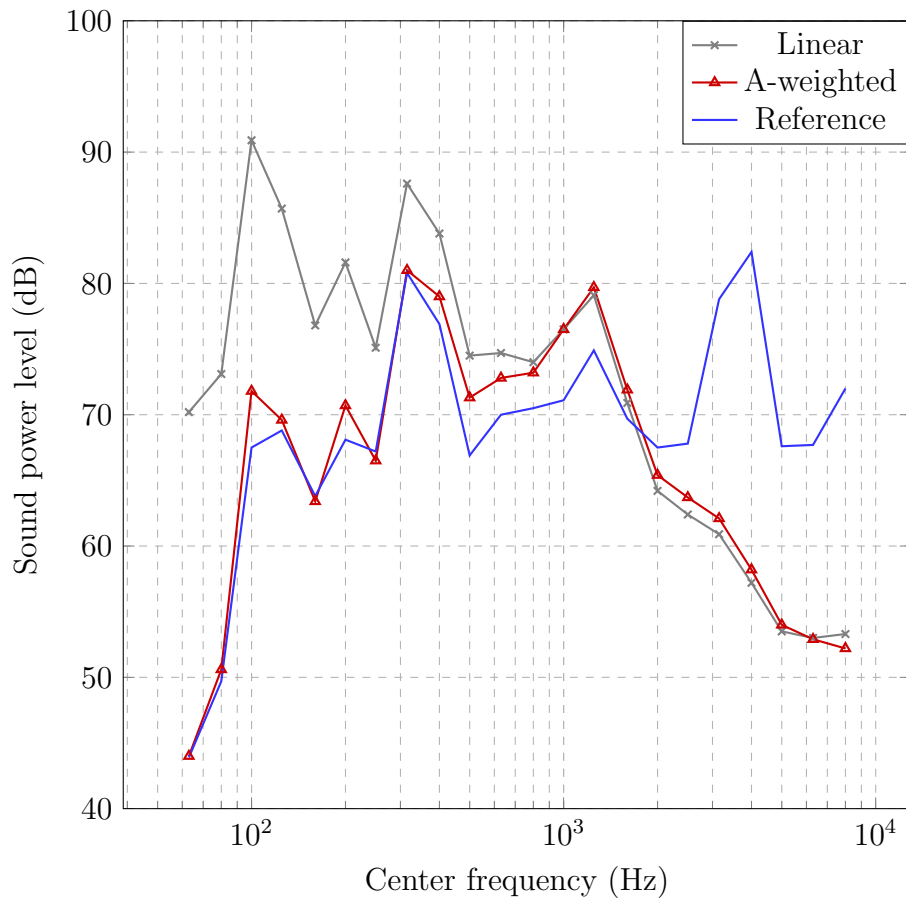


Figure 5.10: Test 6. Sound power level spectrum in one-third octave bands.

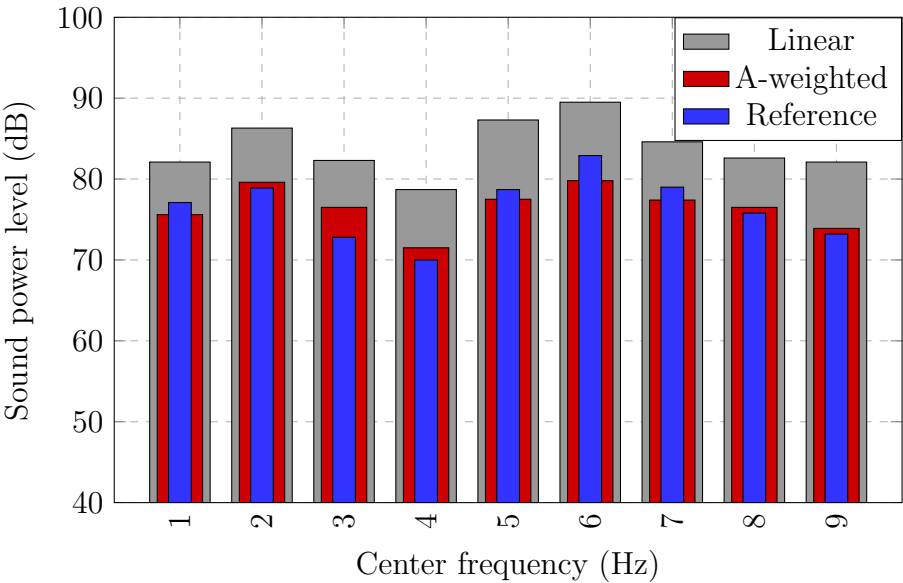


Figure 5.11: Test 6. Sound power level per segment.

Chapter 6

Discussion

There are several points that differ from the initial scope. This is quite expected; it is natural for the scope of a master's thesis to transform. In the initial scope, we were interested to see what means there are available to noise control. These means are discussed in section 3, but we did not have the time to extensively incorporate them to actual measurements. Some of these noise control methods are worth studying in more detail, and many of them could have applications in drives. The main complicating factor is the lack of additional space inside of the drive, and precautions must be taken to maintain acceptable air flow. To tackle this problem, micro perforated plates or acoustic metamaterials could be installed on the inner surface of the cabinet due to their smaller thickness, but more extensive studies are needed to determine the applicable material. Coupled panels with a cavity within the structure are efficient at reducing the noise level, but may require extensive modifications on the cabinet. Another aspect that was in the initial scope was the simulation of a noise source. However, the topic is very broad, and it was deemed too complex a subject with only marginal benefits.

The noise-suppressing grill design proved to be quite an efficient solution. However, it was not examined whether proper air flow is maintained in continuous operation. There are some improvements that can be done on the design, including implementing IP54 filtering and streamlining the design to eliminate discontinuities to further reduce the noise level.

Furthermore, it would have been interesting to compare the two standards, ISO 9614-2:1996 and ISO 3774 for determining sound power level in practice. This was limited by the size of the hemi-anechoic room at disposal. The idea of measuring the noise radiated by a high-power drive is quite novel; we noticed quite late that the methods introduced in the standard ISO 3744 are not suitable. The dimensions of the hemi-anechoic room are too small to incorporate a hemispherical measurement surface, and the measurement

system installed in the room was not fitted to support additional pressure microphones as required by the parallelepiped measurement surface. For these reasons, we decided to use the scanning method.

During this thesis, there were several setbacks that hindered the project. We had issues with installing the drive in the hemi-anechoic room as the room was not designed to incorporate a drive. For future noise measurements of high-power drives, it is important to keep an eye on the temperature as it can change rapidly. Frequent breaks were required in between of scans in order to cool down the frequency drive. In addition, the test cases required that the frequency converters would be running with a load. To achieve this, we fitted the frequency converters to drive a 1 MW motor and a 2 MW motor.

The measurement equipment (the probe and the measurement software license) was not fully at our disposal during the time of the measurements. Same applied to the pressure microphone calibrators. The probe kit that we used was quite dated and did not include a dedicated sound intensity analyzer. Although the pressure microphones and pre-amplifiers served their purpose well, there were some disadvantages to this setup. Exporting the data from the software and post-processing is a tedious task that takes time from the actual analysis of the results. The setup itself is not very mobile as it requires a workstation. To tackle these issues, a dedicated sound intensity analyzer could prove beneficial.

Conducting the measurement can be a lengthy process, especially if it includes a wide frequency range. In such cases, several spacers are needed to increase accuracy at certain frequencies. If the EUT is geometrically complex or does not radiate noise uniformly, additional segments might be needed. Both issues contribute to the total time required by the measurement. The noise measurement of the air-cooled drive, for example, was quite a lengthy process. To obtain reliable results, 78 individual scans (39 per spacer) were required in total per test case. Initially, additional failed scans almost doubled the number of scans.

The scanning method proved quite practical in the end. It provides reliable data as long as some crucial variables are taken into account. These variables include the previous experience of the probe user, reflecting surfaces, extraneous sources, and most importantly the use of correct spacers. Although unreliable data can be detected based on the field indicators, the field indicators are calculated after the measurements have been conducted with the current setup. We understood how problematic this issue was after the measurements for the chronologically first test case (test case 2) were conducted. The main advantage of the method is that it can be used in field conditions, which means that a hemi-anechoic room is not a necessity. As the installation of a drive can easily take a day or two, it could be sensible

to conduct noise tests parallel to other tests for the drive.

Considering previously mentioned advantages and disadvantages, I would recommend investing in a modern sound intensity probe if noise tests are to be conducted repeatedly. There are some major advantages to the investment. Most importantly, the probe kit and the calibrators would be available when needed. Secondly, the lead time of the measurement process and invested man-hours can be dramatically reduced with modern sound intensity probe kits. Some modern probe kits exist that allow measurements that conform to the standard with only one spacer over an extended frequency range of interest, which halves the number of individual scans. For example, the sound analyzer Nor150 manufactured by Norsonic allows sound intensity measurements with only one spacer in the frequency range 25 Hz to 10 kHz; a scenario that typically requires the use of two or three spacers. [26]

Furthermore, the dedicated sound intensity analyzer is able to evaluate the field indicators and warn the user on the go. If the calculated field indicators do not conform to the standard, some analyzers can suggest actions to adjust the scan procedure through graphical user interface, which assists users with little prior experience. For example, the sound analyzer Nor150 fully assists the user throughout the sound intensity measurements conforming to the standard ISO 9614-2. [26] In contrast to the setup used in the thesis, most of the probe kits can be operated by a single person via remote control.

These investments will help with conducting future noise measurements. Although the measurement results give some indication on the actual noise levels produced by the drive, more noise data is needed to generate a more thorough complete picture. I would suggest measuring noise levels of different units with different ratings by selecting the segments in a way that any given segment incorporates only one unit. Practically, this can be achieved only by measuring the unit when it is a part of a lengthier line-up consisting of several units. Obviously, this does not give results with absolute noise levels produced by the unit, as the neighboring units and the position in the line-up may affect the results. However, it can be used to approximate the noise levels produced by a complete line-up.

Chapter 7

Conclusions

The most important finding in this thesis is related to the issue of measuring the noise radiated by a cabinet-built, high-powered frequency converter. As noise tests are also conducted to obtain indicated sound power levels for catalog data, it is important that the test procedure follows a standard that is widely accepted in the industry. The standard ISO 9614-2:1996 introduces a method to determine sound power level that has various advantages over the standard ISO 3744. The method allows sound intensity measurements that conform to the standard also in field conditions with extraneous noise sources. The test cases proved that it produces constantly reliable data given that correct spacers are used and the calculated field indicators are not neglected. As a result, testing procedure and instructions were generated incorporating the standard ISO 9614-2:1996 for any cabinet-built high-powered drive. Also, further steps in measurement instrumentation are recommended if noise tests are incorporated to the standard test protocol.

The measurement procedure is a side product from the measurements conducted for the several predefined test cases. Some of the most interesting aspects in the results of these test cases include the differences in sound power level spectra radiated by the air-cooled drive and the liquid-cooled drive. We found that tonal components can drastically increase the sound power level. Furthermore, we discuss how A-weighting affects the radiated noise spectra, and how these results should be analyzed for frequency converters.

Additionally, several interesting noise control methods based on absorbing materials were discovered. However, these noise control methods should be further studied as there are many factors that determine how viable a method is. This includes the pressure drop it might cause, which might obstruct the air flow inside of the cabinet. For example, if too thick layers of noise absorbing materials are used, the pressure drop can cause the drive to trip as the cooling efficiency is reduced. The optimal method and material both

economically and practically needs extensive evaluation before incorporating to a drive design. The safety features of a material (fire resistivity and non-toxicity) may further increase the costs, hence introducing some of these methods to the standard design might not be economically viable.

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